Schuyler B Serviss
June 6, 1905.

1

THE STEAM-ENGINE

AND

OTHER HEAT-ENGINES.

London: C. J. CLAY AND SONS,
CAMBRIDGE UNIVERSITY PRESS WAREHOUSE,
AVE MARIA LANE.
Glasgow: 50, WELLINGTON STREET.

Leipzig: F. A. BROCKHAUS.
New York: THE MACMILLAN COMPANY.
Bombay and Calcutta: MACMILLAN AND CO., Ltd.

Schuyler B Serviss
June 5, 1905

THE STEAM-ENGINE

AND

OTHER HEAT-ENGINES

BY

James Alfred

J. A. EWING, M.A., LL.D., F.R.S., M.INST.C.E.,

PROFESSOR OF MECHANISM AND APPLIED MECHANICS IN
THE UNIVERSITY OF CAMBRIDGE.
FELLOW OF KING'S COLLEGE, CAMBRIDGE.

STEREOTYPED EDITION.

CAMBRIDGE:
AT THE UNIVERSITY PRESS.
1902

First Edition 1894.
Second Edition, revised 1897.
Reprinted 1899, 1902.

PREFACE TO THE FIRST EDITION.

WHEN I undertook some years ago to prepare an article on the Steam-Engine and other Heat-Engines for the *Encyclopædia Britannica* it seemed that the subject might be appropriately treated by following the general lines which had been found suitable in lecturing to students of engineering. The article was accordingly written on these lines, but necessarily in a very condensed form.

From the time of its publication I have hoped to expand it into a University Text-book, and have been encouraged by more than one other teacher to believe that such a text-book might be useful. The present work is the outcome of that intention: it is based on the *Encyclopædia* article, but the additions and changes have been so considerable that except for parts of one or two Chapters the book is virtually new.

The design has been to treat not only of the thermodynamics of the steam-engine, but of other aspects of the subject which admit of theoretical discussion, such as the kinematics of the slide-valve and the kinetics of the governor and of the moving mechanism as a whole, and also to give a general, if brief, account of the forms taken by actual engines and of the manner of their working. No attempt has been made to describe details particularly, but the distinguishing features of certain types have been indicated. In doing this the greatest amount of space has been given to the less familiar forms, on the principle that a student need be at no loss to learn the construction of engines of the commoner kinds. Air, gas, and oil engines are noticed, as well as steam-engines.

The endeavour throughout has been to make evident the bearing of theory on practical issues. The experimental study

of steam-engines, which has done much to bring thermodynamics into closer contact with engineering, is described at some length. It is now usual for students to combine their lecture-room study of heat-engines with work in the laboratory as well as in the drawing office, and parts of the book are designed to serve in some measure as a manual for the steam laboratory.

The Chapters which relate to applied thermodynamics cannot of course pretend to give so full a treatment as will be found in the valuable books of Professor Cotterill and Professor Peabody, which are devoted entirely to that subject. The theory of heat-engines is presented here in small compass and in elementary form, but I hope it may be found that this section is, so far as it goes, complete, and that there are not many serious omissions in regard to matters of practical importance. I have made considerable use of the entropy-temperature diagram as a means of exhibiting thermodynamic actions, believing that this construction only requires to be better known to be widely appreciated by engineers. In calculations affected by the mechanical equivalent of heat, 778 foot-pounds has been taken as the value of Joule's equivalent, the recent work of Griffiths and Rowland, in conjunction with the later researches of Joule himself, having left no doubt that this number or one closely approximating to it is to be accepted in place of the familiar 772.

I have to thank Messrs A. and C. Black, the publishers of the *Encyclopædia Britannica*, for consenting to an arrangement which allowed some of the material of the article "Steam-Engine" to be utilized, and also my present and former assistants, Mr W. E. Dalby and Mr T. Reid, for much kind help in preparing illustrations. I am indebted for drawings to Messrs Galloway, Messrs Gourlay, Messrs Thornycroft, Messrs Willans and Robinson, Mr H. Davey, Dr Kirk, Mr C. A. Parsons, Mr F. W. Webb, and other engineers.

J. A. EWING.

ENGINEERING LABORATORY, CAMBRIDGE.
April 25, 1894.

PREFACE TO THE SECOND EDITION.

IN the Second Edition the book has been revised throughout, and a considerable amount of new matter has been added. In particular the section relating to gas-engines has been extended. For suggesting several of the other additions and emendations I am indebted to my friends and former assistants, Professor Nicolson, of McGill University, Montreal, Professor Dalby, of the Finsbury Technical College, and Professor Dunkerley, of the Royal Naval College, Greenwich.

J. A. EWING.

CAMBRIDGE,
December, 1897.

CONTENTS.

CHAPTER I.

THE EARLY HISTORY OF THE STEAM-ENGINE.

CHAPTER II.

ELEMENTARY THEORY OF HEAT-ENGINES.

CHAPTER III.

PROPERTIES OF STEAM AND ELEMENTARY THEORY OF THE STEAM-ENGINE.

CHAPTER IV.

FURTHER POINTS IN THE THEORY OF HEAT-ENGINES.

CHAPTER V.

ACTUAL BEHAVIOUR OF STEAM IN THE CYLINDER.

CHAPTER VI.

THE TESTING OF STEAM-ENGINES.

CHAPTER VII.

COMPOUND EXPANSION.

CHAPTER VIII.

VALVES AND VALVE-GEARS.

CHAPTER IX.

GOVERNING.

CHAPTER X.

THE WORK ON THE CRANK-SHAFT.

CHAPTER XI.

THE PRODUCTION OF STEAM.—BOILERS.

CHAPTER XII.

FORMS OF THE STEAM-ENGINE.

CHAPTER XIII.

AIR-ENGINES.

CHAPTER XIV.

GAS-ENGINES AND OIL-ENGINES.

CHAPTER I.

THE EARLY HISTORY OF THE STEAM-ENGINE.

1. **Heat-Engines in general.** In the scientific treatment of the steam-engine we have in the first place, and mainly, to regard it as a heat-engine—that is, a machine in which heat is employed to do mechanical work. Other aspects of the steam-engine will present themselves when we come to examine the action of the mechanism in detail, but the foremost place must be given to thermodynamic considerations. From the thermodynamic point of view the function of a heat-engine is to get as much work as possible from a given supply of heat, or (to go a step further back) from the combustion of a given quantity of fuel. Hence a large part of our subject is the discussion of what is called the *efficiency* of the engine, which is the ratio of the work done to the heat supplied. We have to consider on what conditions efficiency depends, how its value is limited in theory and how nearly the limiting value may be attained in practice. We have to describe means of testing the efficiency of engines, and the results which such tests have given in actual cases. Much of what has to be said in regard to efficiency is applicable to all heat-engines, whatever be the character of the substance which is made use of as the means of doing work within the engine. In all practical heat-engines work is done through the expansion by heat of a fluid which exerts pressure and overcomes resistance as it expands. Thus in steam-engines the working substance is water and water-vapour, and work is done by the pressure which the substance exerts while its volume is undergoing change. In air-engines the working substance is

atmospheric air; in gas-engines and oil-engines it is a mixture of air with combustible gas or vaporised oil and with the products of combustion. These last are important instances of what are sometimes called internal combustion engines, in which the heat is developed by combustion occurring within the working substance itself instead of reaching the substance from an external source. We may have heat-engines in which the working substance is not a fluid, and examples might even be named in which a substance that is alternately heated and cooled is made to do work not in consequence of changes in its volume or in its form, but in consequence of some other effect of heat such, for instance, as the loss and gain of magnetic quality. A complete list of typical heat-engines would include a mention of guns, in which the heat that is generated by the combustion of an explosive does work in giving energy of motion to a projectile. We have, however, to do only with those types of heat-engine whose object is to change the potential energy of fuel into a manageable mechanical form, so that they may serve as prime movers to other mechanism. Of such engines the steam-engine is by far the most important.

As a preliminary to the study of the modern engine it will be useful to review, if only very briefly, some of the stages through which it has passed in its development. In any such historical sketch the largest share of attention necessarily falls to the work of Watt, whose inventions were as remarkable for their scientific interest as for their industrial importance. But it should be borne in mind that a process of evolution had been going on before the time of Watt which prepared the steam-engine for the immense improvements it received at his hands. The labours of Watt stand in a natural sequence to those of Newcomen, and Newcomen's to those of Papin and Savery. Savery's engine, again, was the reduction to practical form of a contrivance which had long before been known as a scientific toy.

2. **Hero of Alexandria.** The earliest notices of heat-engines are found in the *Pneumatics* of Hero of Alexandria, which dates from the second century before Christ. One of the contrivances mentioned there is the æolipile, a steam reaction-turbine consisting of a spherical vessel pivoted on a central axis and supplied with steam through one of the pivots. The steam

escapes by bent pipes facing tangentially in opposite directions, at opposite ends of a diameter perpendicular to the axis. The globe revolves by reaction from the escaping steam, just as a Barker's mill is driven by escaping water. Another apparatus described by Hero (fig. 1)[1] is interesting as the prototype of a class of engines which long afterwards became practically important. A hollow altar containing air is heated by a fire kindled on it; the air in expanding drives some of the water contained in a spherical vessel beneath the altar into a bucket, which descends and opens the temple doors above by pulling round a pair of vertical posts to which the doors are fixed. When the fire is extinguished the air cools, the water leaves the bucket, and the doors close. In another device a jet of water driven out by expanding air is turned to account as a fountain. Several other philosophical toys or pieces of conjuring apparatus of the like kind are also described, but there is no suggestion that the methods they illustrate could be applied on a large scale or turned to any useful account.

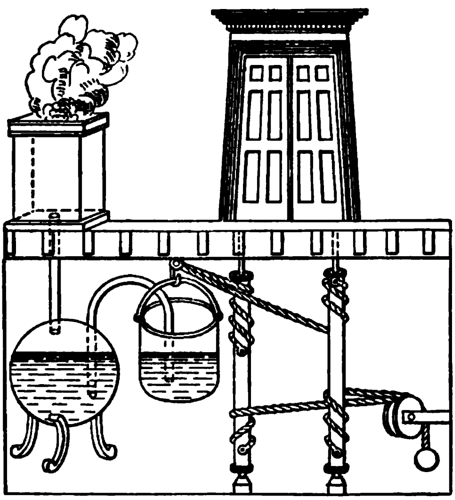


FIG. 1. Apparatus described by Hero.

3. Della Porta and De Caus. From the time of Hero to the 17th century there is no progress to record, though here

[1] From Greenwood's translation of Hero's *Pneumatics*, edited by B. Woodcroft, 1851.

and there we find evidence that appliances like those described by Hero were used for trivial purposes, such as organ-blowing and the turning of spits. The next distinct step was the publication in 1606 of a treatise on pneumatics by Giovanni Battista Della Porta, in which he shows an apparatus similar to Hero's fountain, but with steam instead of air as the displacing fluid. Steam generated in a separate vessel passed into a closed chamber containing water, and drove the water out through a pipe which opened near the bottom of the vessel. He also points out that the condensation of steam in the closed chamber may be used to produce a vacuum and suck up water from a lower level. In fact, his suggestions anticipate very fully the principle which a century later was applied by Savery in the earliest commercially successful steam-engine. In 1615 Salomon De Caus gives a plan of forcing up water by a steam-fountain which differs from Porta's only in having one vessel serve both as boiler and as displacement-chamber, the hot water being itself raised.

4. **Branca's Steam Turbine.** Another line of invention was taken by Giovanni Branca (1629), who designed an engine shaped like a water-wheel, to be driven by the impact of a jet of steam on its vanes, and, in its turn, to drive other mechanism for various useful purposes. But Branca's suggestion was unproductive, and we find the course of invention revert to the line followed by Porta and De Caus.

5. **Marquis of Worcester.** The next contributor is one whose place is not easily assigned. To Edward Somerset, second marquis of Worcester, appears to be due the credit of proposing, if not of making, the first useful steam-engine. Its object was to raise water, and it worked probably like Porta's model, but with a pair of displacement-chambers, from each of which alternately water was forced by steam from an independent boiler, or perhaps by applying heat to the chamber itself, while the other vessel was allowed to refill. The only description of the engine is found in Art. 68 of Worcester's *Century of Inventions* (1663). There are no drawings, and the notice is so obscure that it is difficult to say whether there were any distinctly novel features except the double action. The inventor's account leaves much to the imagination. It is entitled "A Fire Water-work," and runs thus:—

"An admirable and most forcible way to drive up water by fire, not by drawing or sucking it upwards, for that must be as the Philosopher calleth it, *Intra sphaeram activitatis*, which is but at such a distance. But this way hath no Bounder, if the Vessels be strong enough; for I have taken a piece of a whole Cannon, whereof the end was burst, and filled it three-quarters full of water, stopping and scruing up the broken end; as also the Touch-hole; and making a constant fire under it, within 24 hours it burst and made a great crack. So that having a way to make my Vessels, so that they are strengthened by the force within them, and the one to fill after the other, I have seen the water run like a constant Fountaine-stream forty foot high; one Vessel of water rarified by fire driveth up forty of cold water. And a man that tends the work is but to turn two Cocks, that one Vessel of water being consumed, another begins to force and re-fill with cold water, and so successively, the fire being tended and kept constant, which the self-same Person may likewise abundantly perform in the interim between the necessity of turning the said Cocks."

Later articles in the *Century of Inventions* contain notices of a device which under the name of a "Water-commanding Engine" received protection by Act of Parliament and was experimented on by Worcester on a large scale at Vauxhall. But there is nothing to show distinctly that the Water-commanding Engine was a heat-engine at all, and the meagre accounts that have been given of it rather point to the conclusion that it was a form of "Perpetual Motion." In any case the experiments led to no practical result.

6. **Savery.** The steam-engine became commercially successful in the hands of Thomas Savery, who in 1698 obtained a patent for a water-raising engine, shown in fig. 2. Steam is admitted to one of the oval vessels A, displacing water, which it drives up through the check-valve B. When the vessel A is emptied of water, the supply of steam is stopped, and the steam already there is condensed by allowing a jet of cold water from a cistern above to stream over the outer surface of the vessel. This produces a vacuum and causes water to be sucked up through the pipe C and the valve D. Meanwhile, steam has been displacing water from the other vessel, and is ready to be condensed there. The valves B and D open only upwards. The supplementary boiler and furnace E are for feeding water to the main boiler; E is filled while cold and a fire is lighted under it; it then acts like the vessel of De Caus in forcing a supply of feed-water into the main boiler F. The gauge-cocks G, G for testing the level of the water in the boiler are an interesting feature of detail. Another form of Savery's engine had only one displacement-chamber and worked intermittently. In the use of artificial means to condense the

steam, and in the application of the vacuum so formed to raise water by suction from a level lower than that of the engine, the action used by Savery was probably an advance on that proposed or used by Worcester; in any case Savery's was the first engine to

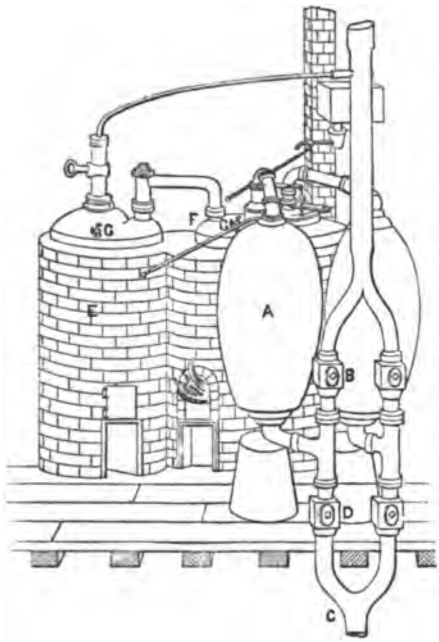


FIG. 2. Savery's Pumping Engine, 1698.

take a really practical shape. It found considerable employment in pumping mines and in raising water to supply houses and towns, and even to drive water-wheels. A serious difficulty which prevented its general use in mines was the fact that the height through which it would lift water was limited by the pressure the boiler and vessels could bear. Pressures as high as 8 or 10 atmospheres were employed—and that, too, without a safety-valve. But Savery found it no easy matter to deal with high-pressure steam; he complains that it melted his common solder, and forced him, as Desaguliers tells us, "to be at the pains and charge to have all his joints soldered with spelter." Apart from this drawback the waste of fuel was enormous, from the condensation of steam which took place on the surface of the water and on the sides of the displacement-chamber at each stroke; the consumption of coal was, in proportion to the work done, some twenty times

greater than it is in a good modern steam-engine. In a tract called *The Miner's Friend*, Savery alludes thus to the alternate heating and cooling of the water-vessel: "On the outside of the vessel you may see how the water goes out as well as if the vessel were transparent, for so far as the steam continues within the vessel so far is the vessel dry without, and so very hot as scarce to endure the least touch of the hand. But as far as the water is, the said vessel will be cold and wet where any water has fallen on it; which cold and moisture vanishes as fast as the steam in its descent takes place of the water." Before Savery's engine was entirely displaced by its successor, Newcomen's, it was improved by Desaguliers, who applied to it the safety-valve (invented by Papin), and substituted condensation by a jet of cold water within the vessel for the surface condensation used by Savery.

To Savery is ascribed the first use of the familiar term "horse-power" as a measure of the performance of an engine.

7. Gunpowder Engines. Some twenty years before the date of Savery's patent, proposals had been made by several inventors to raise water by means of the explosive power of gunpowder. One scheme was to explode the powder in a closed vessel furnished with valves which opened outwards and allowed a great part of the air and burnt gases to escape when the explosion took place. As the gas that remained became cool a partial vacuum was formed in the vessel, and this was used to draw up water from a lower level. It does not appear that these schemes were ever put in practice except experimentally. The most interesting of the gunpowder engines was that of Huygens (1680), who for the first time introduced the piston and cylinder as constituent parts of a heat-engine. In Huygens' engine the piston was set at the top of a vertical cylinder and a charge of powder was exploded below it. This expelled part of the gaseous contents through valves which opened outwards, and then the cooling of the remainder caused the piston to descend under atmospheric pressure. The piston in descending did work by raising a weight through the medium of a cord and pulley.

8. Papin. In 1690 Denis Papin, who ten years before had invented the safety-valve as an adjunct to his "digester," suggested that the condensation of steam should be employed to make a vacuum under a piston which had been previously raised by the

expansion of the steam. Papin had been associated with Huygens in his experiments on the production of a vacuum under a piston by means of gunpowder, and had described Huygens' machine to the Royal Society. Noticing that after the explosion enough gas remained in the cylinder to fill about one-fifth of its volume, after cooling, he cast about for some means of obtaining a better vacuum. "By another way, therefore, I endeavoured to attain the same end, and since it is a property of water that a small quantity of it, converted into steam by heat, has an elastic force like that of air, but when cold supervenes, is again resolved into water so that no trace of the said elastic force remains, I saw that machines might be constructed wherein water, by means of no very intense heat and at small cost, might produce that perfect vacuum which had failed to be obtained by the use of gunpowder." He goes on to describe what was unquestionably the earliest cylinder and piston steam-engine, and his plan of

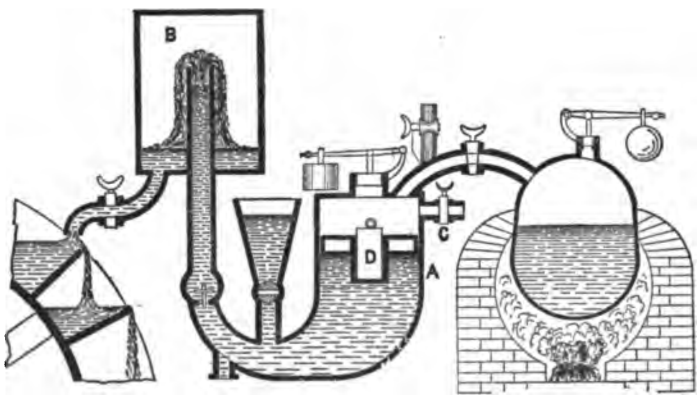


FIG. 3. Papin's modification of Savery's Engine, 1705.

using steam was that which afterwards took practical shape in the atmospheric engine of Newcomen. But his scheme was made unworkable by the fact that he proposed to use but one vessel as both boiler and cylinder. A small quantity of water was placed at the bottom of a cylinder and heat was applied. When the piston had risen the fire was removed, the steam was allowed to cool, and the piston did work in its down-stroke under the pressure of the atmosphere.

After hearing of Savery's engine in 1705 Papin turned his attention to improving it, and devised a modified form, shown

in fig. 3, in which the displacement-chamber A was a cylinder, with a floating diaphragm or piston on the top of the water to keep the water and steam from direct contact with one another. The water was delivered into a closed air-vessel B, from which it issued in a continuous stream against the vanes of a water-wheel. After the steam had done its work in the displacement-chamber it was allowed to escape by the stop-cock C instead of being condensed. This second engine of Papin's was in fact a non-condensing single-acting steam-pump, with steam-cylinder and pump-cylinder in one. A curious feature of it was the heater D, a mass of hot metal placed in the diaphragm for the purpose of keeping the steam dry. Among the many inventions of Papin was a boiler with an internal fire-box,—the earliest example of a construction that is now almost universal[1].

9. Newcomen's "Atmospheric" Engine. While Papin was thus going back from his first notion of a piston-engine to Savery's cruder type, a new inventor had appeared who made the piston-engine a practical success by separating the boiler from the cylinder and by using (as Savery had done) artificial means to condense the steam. This was Newcomen, who in 1705, in conjunction with Savery and with Cawley, gave the steam-engine the form shown in fig. 4. The piston was connected by a chain with one end of an overhead beam. Steam admitted from the boiler to the cylinder allowed the piston to be raised by a heavy counterpoise hanging from the beam near the other end. Then the steam-valve was shut and a jet of cold water entered the cylinder and condensed the steam. The piston was consequently forced down by the pressure of the atmosphere and did work on the pump through the medium of a long rod which hung from the other end of the beam. The next entry of steam expelled the condensed water from the cylinder through an escape valve. The piston was kept tight by a layer of water on its upper surface. Condensation was at first effected by cooling the outside of the cylinder, but an accidental leakage of the packing water past the piston showed the advantage of condensing by a jet of injection water, and this plan took the place of surface condensation. The engine used steam which had a pressure little if at all greater

[1] For an account of Papin's inventions, see his *Life, and Correspondence with Leibnitz and Huygens*, by Dr E. Gerland, Berlin, 1881. See also Muirhead's *Life of Watt*.

than that of the atmosphere; sometimes indeed it was worked with the manhole-lid off the boiler. The function of the steam

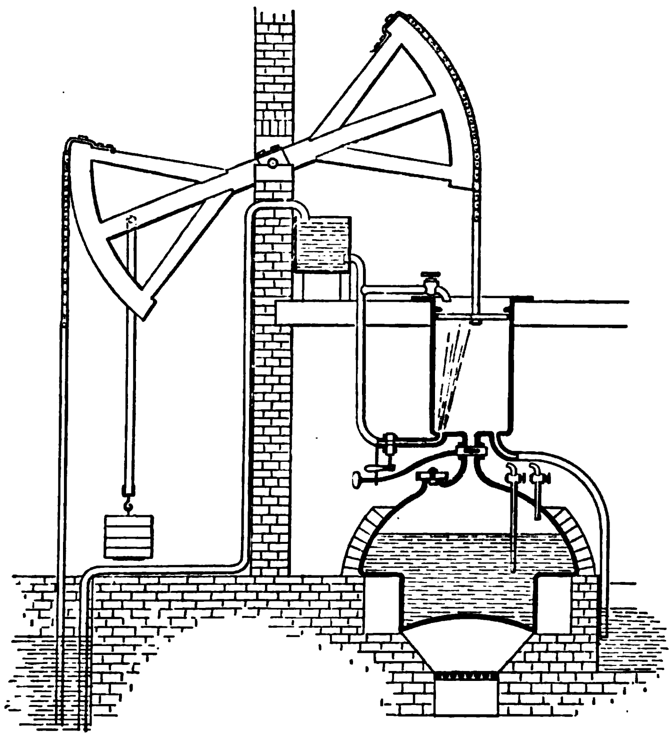


FIG. 4. Newcomen's Atmospheric Engine, 1705.

was merely to allow the piston to be raised, by making the pressure on the under side equal or nearly equal to the pressure on the top, and then to produce a vacuum by being condensed. Newcomen's engine was essentially the cylinder and piston of Papin combined with the separate boiler of Savery.

About 1711 Newcomen's engine began to be introduced for pumping mines. It is doubtful whether the engine was originally automatic in its action or depended on the periodical turning of taps by an attendant. An old print of an engine erected by Newcomen in 1712 near Dudley Castle shows a species of automatic gear. The common story is that in 1713 a boy named Humphrey Potter, whose duty it was to open and shut the valves of an engine he attended, made the engine self-acting by causing the beam itself to open and close the valves by means of cords

and catches. This rude device was simplified in 1718 by Henry Beighton, who suspended from the beam a rod called the plug-tree, which worked the valves by means of tappets. By 1725 the engine was in common use in collieries, and it held its place without material change for about three-quarters of a century in all. Near the close of its career the atmospheric engine was much improved in its mechanical details by Smeaton, who built many large engines of this type about the year 1770, just after the great step which was to make Newcomen's engine obsolete had been taken by James Watt.

Like Savery's engine, Newcomen's was put to no other use than to pump water—in some instances for the purpose of turning water-wheels to drive other machinery. Compared with Savery's it had the great advantage that the intensity of pressure in the pump was not in any way limited by the pressure of the steam, but could be made as great as might be desired by reducing the area of the pump plunger. It shared with Savery's, in a scarcely less degree, the defect already pointed out, that steam was wasted by the alternate heating and cooling of the vessel into which it was led. Even contemporary writers complain of its "vast consumption of fuel," which appears to have been scarcely smaller than that of the engine of Savery.

10. James Watt. In 1763 James Watt, an instrument maker in Glasgow, while engaged by the University in repairing a model of Newcomen's engine, was struck with the waste of steam to which the alternate chilling and heating of the cylinder gave rise. He saw that the remedy, in his own words, would lie in keeping the cylinder as hot as the steam that entered it. With this view he added to the engine a new organ—namely, the *condenser*—a vessel separate from the cylinder, into which the steam should be allowed to escape from the cylinder, to be condensed there by the application of cold water either outside or as a jet. To preserve the vacuum in his condenser he added a pump, called the air-pump, whose function was to pump from it the condensed steam and water of condensation, as well as the air which would otherwise accumulate by leakage inwards or by being brought in with the steam or with the injection water. Then as the cylinder was itself no longer used as the chamber in which the steam was condensed he was able to keep it continuously hot by

clothing it with non-conducting bodies, and in particular by the use of a *steam-jacket*, or layer of hot steam between the cylinder and an external casing. Further, and still with the same object, he covered in the top of the cylinder, taking the piston-rod out through a steam-tight stuffing-box, and allowed steam instead of air to press upon the piston's upper surface. The idea of using a separate condenser had no sooner occurred to Watt than he put it to the test by constructing the apparatus shown in fig. 5. There A is the cylinder, B a condenser (of the type now distinguished as a surface-condenser) and C is the air-pump. The cylinder was filled with steam above the piston, and a vacuum was formed in the surface-condenser B. On opening the stop-cock D the steam rushed over from the cylinder and was condensed, while the piston rose and lifted a weight. A fuller account of this experiment will be found in Watt's narrative, below.

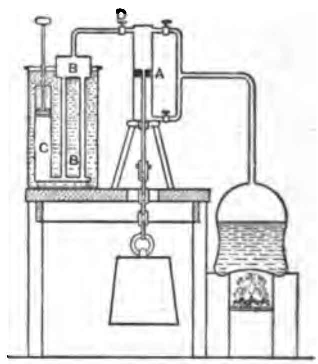


FIG. 5.
Watt's Experimental Apparatus.

After several trials Watt patented his improvements in 1769; they are described in his specification in the following words, which, apart from their immense historical interest, deserve careful study as a statement of principles which to this day guide the scientific development of the steam-engine :—

"My method of lessening the consumption of steam, and consequently fuel, in fire-engines, consists of the following principles:—

"*First*, That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire-engines, and which I call the steam-vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

"*Secondly*, In engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam-vessels or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water or other cold bodies.

"*Thirdly*, Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the

steam-vessels or condensers by means of pumps, wrought by the engines themselves, or otherwise.

"*Fourthly*, I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in common fire-engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the open air after it has done its office......

"*Sixthly*, I intend in some cases to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably, so that the engines shall be worked by the alternate expansion and contraction of the steam.

"*Lastly*, Instead of using water to render the pistons and other parts of the engine air and steam-tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver and other metals in their fluid state."

The fifth claim was for a rotary engine, and need not be quoted here.

The "common fire-engine" alluded to was the steam-engine, or, as it was more generally called, the "atmospheric" engine, of Newcomen. Enormously important as Watt's first patent was, it resulted for a time in the production of nothing more than a greatly improved engine of the Newcomen type, much less wasteful of fuel, able to make faster strokes, but still only suitable for pumping, still single-acting, with steam admitted during the whole stroke, the piston still pulling the beam by a chain working on a circular arc. The condenser was generally kept cool by the injection of cold water, but Watt has left a model of a surface-condenser made up of small tubes, in every essential respect like the condensers now used in marine engines. He also used, as we have seen, a surface-condenser in the experimental apparatus by which the practicability of condensation in a separate vessel was first demonstrated.

11. Watt's pumping-engine of 1769. Fig. 6 is an example of the Watt pumping-engine of this period. It should be noticed that, although the top of the cylinder is closed and steam has access to the upper side of the piston, this is done only to keep the cylinder and piston warm. The engine is still single-acting; the steam on the upper side merely plays the part which was played in Newcomen's engine by the atmosphere; and it is the lower end of the cylinder alone that is ever put in communication with the condenser. There are three valves,—the "steam" valve *a*, the "equilibrium" valve *b*, and the "exhaust"

valve *c*. At the beginning of the down-stroke *c* is opened to produce a vacuum below the piston and *a* is opened to admit steam above it. At the end of the down-stroke *a* and *c* are shut

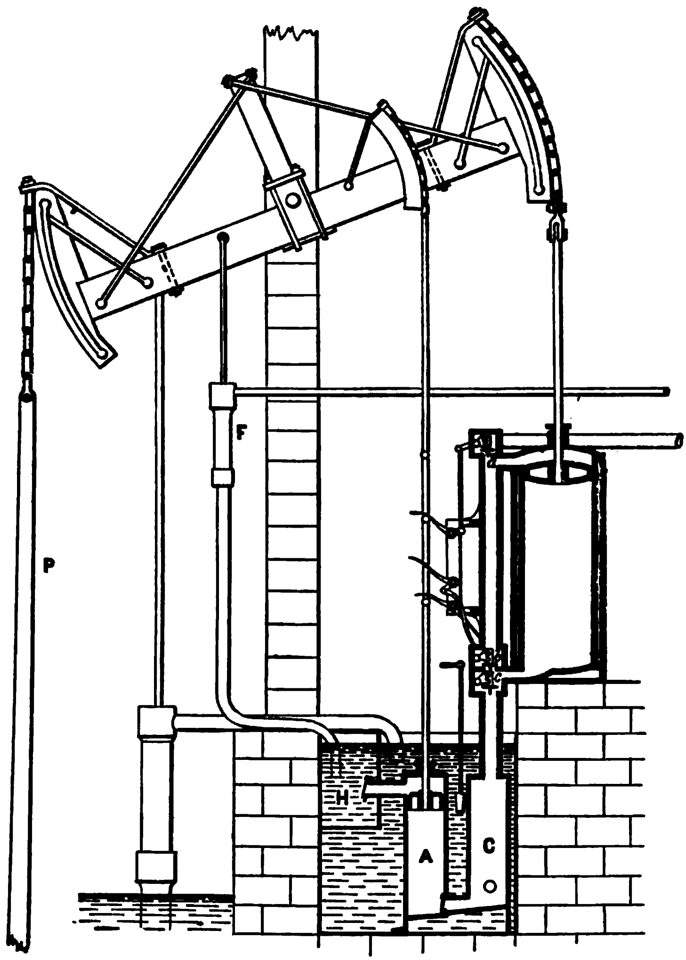


Fig. 6. Watt's Single-acting Engine, 1769.

and *b* is opened. This puts the two sides of the piston in equilibrium, and allows the piston to be pulled up by the pump-rod P, which is heavy enough to serve as a counterpoise. C is the condenser, and A the air-pump, which discharges into the hot well H, whence the supply of the feed-pump F is drawn.

12. Watt's narrative of his invention. In a note appended to the article "Steam-Engine" in Robison's *System of Mechanical Philosophy* (1822) Watt has given the following account of the experiments and reflexions which led up to his first patent. This narrative is of so particular interest that no apology need be made for reproducing it in full.

"My attention was first directed in the year 1759 to the subject of steam-engines, by the late Dr Robison himself, then a student in the University of Glasgow, and nearly of my own age. He at that time threw out an idea of applying the power of the steam-engine to the moving of wheel-carriages, and to other purposes, but the scheme was not matured, and was soon abandoned on his going abroad.

"About the year 1761 or 1762, I tried some experiments on the force of steam, in a Papin's digester, and formed a species of steam-engine by fixing upon it a syringe one-third of an inch diameter, with a solid piston, and furnished also with a cock to admit the steam from the digester, or shut it off at pleasure, as well as to open a communication from the inside of the syringe to the open air, by which the steam contained in the syringe might escape. When the communication between the digester and syringe was opened, the steam entered the syringe, and by its action upon the piston raised a considerable weight (15 lbs.) with which it was loaded.

"When this was raised as high as was thought proper, the communication with the digester was shut, and that with the atmosphere opened; the steam then made its escape, and the weight descended. The operations were repeated, and though in this experiment the cock was turned by hand, it was easy to see how it could be done by the machine itself, and to make it work with perfect regularity. But I soon relinquished the idea of constructing an engine upon this principle, from being sensible it would be liable to some of the objections against Savery's engine, viz. the danger of bursting the boiler, and the difficulty of making the joints tight, and also that a great part of the power of the steam would be lost, because no vacuum was formed to assist the descent of the piston. [I, however, described this engine in the fourth article of the specification of my patent of 1769; and again in the specification of another patent in the year 1784, together with a mode of applying it to the moving of wheel-carriages.]

"The attention necessary to the avocations of business prevented me from then prosecuting the subject farther; but in the winter of 1763–4, having occasion to repair a model of Newcomen's engine belonging to the Natural Philosophy class of the University of Glasgow, my mind was again directed to it. At that period, my knowledge was derived principally from Desaguliers, and partly from Belidor. I set about repairing it as a mere mechanician, and when that was done and it was set to work, I was surprised to find that its boiler could not supply it with steam, though apparently quite large enough (the cylinder of the model being two inches in diameter and six inches stroke, and the boiler about nine inches diameter). By blowing the fire it was made to take a few strokes, but required an enormous quantity of injection water, though it was very lightly loaded by the column of water in the pump. It soon occurred that this was caused by the little cylinder exposing a greater surface to condense the steam than the cylinders of larger engines did in proportion to their respective contents. It was found that by shortening the column of water in the pump, the boiler could supply the cylinder with steam, and

that the engine would work regularly with a moderate quantity of injection. It now appeared that the cylinder of the model being of brass, would conduct heat much better than the cast-iron cylinders of larger engines (generally covered on the inside with a stony crust), and that considerable advantage could be gained by making the cylinders of some substance that would receive and give out heat slowly: of these, wood seemed to be the most likely, provided it should prove sufficiently durable.

"A small engine was therefore constructed with a cylinder six inches diameter, and twelve inches stroke, made of wood, soaked in linseed oil, and baked to dryness. With this engine many experiments were made; but it was soon found that the wooden cylinder was not likely to prove durable, and that the steam condensed in filling it still exceeded the proportion of that required for large engines according to the statements of Desaguliers. It was also found, that all attempts to produce a better exhaustion by throwing in more injection, caused a disproportionate waste of steam. On reflection, the cause of this seemed to be the boiling of water in vacuo at low heats, a discovery lately made by Dr Cullen, and some other philosophers (below 100°, as I was then informed), and, consequently, at greater heats, the water in the cylinder would produce a steam which would, in part, resist the pressure of the atmosphere.

"By experiments which I then tried upon the heats at which water boils under several pressures greater than that of the atmosphere, it appeared, that when the heats proceeded in an arithmetical, the elasticities proceeded in some geometrical ratio; and by laying down a curve from my data, I ascertained the particular one near enough for my purpose. It also appeared, that any approach to a vacuum could only be obtained by throwing in large quantities of injection, which would cool the cylinder so much as to require quantities of steam to heat it again, out of proportion to the power gained by the more perfect vacuum; and that the old engineers had acted wisely in contenting themselves with loading the engine with only six or seven pounds on each square inch of the area of the piston.

"It being evident that there was a great error in Dr Desaguliers' calculations of Mr Beighton's experiments on the bulk of steam, a Florence flask, capable of containing about a pound of water, had about one ounce of distilled water put into it; a glass tube was fitted into its mouth, and the joining made tight by lapping that part of the tube with packthread covered with glazier's putty. When the flask was set upright, the tube reached down near to the surface of the water, and in that position the whole was placed in a tin reflecting oven before a fire, until the water was wholly evaporated, which happened in about an hour, and might have been done sooner had I not wished the heat not much to exceed that of boiling water. As the air in the flask was heavier than the steam, the latter ascended to the top, and expelled the air through the tube.

"When the water was all evaporated, the oven and flask were removed from the fire, and a blast of cold air was directed against one side of the flask, to collect the condensed steam in one place. When all was cold, the tube was removed, the flask and its contents were weighed with care; and the flask being made hot, it was dried by blowing into it by bellows, and when weighed again, was found to have lost rather more than four grains, estimated at 4½ grains.

"When the flask was filled with water, it was found to contain about 17¼ ounces avoirdupois of that fluid, which gave about 1800 for the expansion of water converted into steam of the heat of boiling water.

"This experiment was repeated with nearly the same result; and in order to ascertain whether the flask had been wholly filled with steam, a similar quantity of water was for the third time evaporated; and, while the flask was still cold, it was

placed inverted, with its mouth (contracted by the tube) immersed in a vessel of water, which it sucked in as it cooled, until in the temperature of the atmosphere it was filled to within half-an-ounce measure of water. [In the contrivance of this experiment I was assisted by Dr Black. In Dr Robison's edition of Dr Black's lectures, Vol. I. page 147, the latter hints at some experiments upon this subject as made by him; but I have no knowledge of any except those which I made myself.]

"In repetitions of this experiment at a later date, I simplified the apparatus by omitting the tube, and laying the flask upon its side in the oven, partly closing its mouth by a cork having a notch on one side, and otherwise proceeding as has been mentioned. I do not consider these experiments as extremely accurate, the only scale-beam of a proper size which I had then at my command not being very sensible, and the bulk of the steam being liable to be influenced by the heat to which it is exposed, which, in the way described, is not easily regulated or ascertained; but, from my experience in actual practice, I esteem the expansion to be rather more than I have computed.

"A boiler was constructed which showed, by inspection, the quantity of water evaporated in any given time, and thereby ascertained the quantity of steam used in every stroke by the engine, which I found to be several times the full of the cylinder. Astonished at the quantity of water required for the injection, and the great heat it had acquired from the small quantity of water in the form of steam which had been used in filling the cylinder, and thinking I had made some mistake, the following experiment was tried:—A glass tube was bent at right angles, one end was inserted horizontally into the spout of a tea-kettle, and the other part was immersed perpendicularly in well-water contained in a cylindric glass vessel, and steam was made to pass through it until it ceased to be condensed, and the water in the glass vessel was become nearly boiling hot. The water in the glass vessel was then found to have gained an addition of about one-sixth part from the condensed steam. Consequently, water converted into steam can heat about six times its own weight of well-water to 212°, or till it can condense no more steam. Being struck with this remarkable fact, and not understanding the reason of it, I mentioned it to my friend Dr Black, who then explained to me his doctrine of latent heat, which he had taught for some time before this period (summer 1764), but having myself been occupied with the pursuits of business, if I had heard it I had not attended to it, when I thus stumbled upon one of the material facts by which that beautiful theory is supported.

"On reflecting further, I perceived, that in order to make the best use of steam, it was necessary, first, that the cylinder should be maintained always as hot as the steam which entered it; and secondly, that when the steam was condensed, the water of which it was composed, and the injection itself, should be cooled down to 100°, or lower, where that was possible. The means of accomplishing these points did not immediately present themselves; but early in 1765 it occurred to me, that if a communication were opened between a cylinder containing steam, and another vessel which was exhausted of air and other fluids, the steam, as an elastic fluid, would immediately rush into the empty vessel, and continue so to do until it had established an equilibrium; and if that vessel were kept very cool by an injection or otherwise, more steam would continue to enter until the whole was condensed. But both the vessels being exhausted, or nearly so, how was the injection water, the air which would enter with it, and the condensed steam, to be got out?

"This I proposed, in my own mind, to perform in two ways. One was by adapting to the second vessel a pipe reaching downwards more than 34 feet, by

which the water would descend (a column of that length overbalancing the atmosphere), and by extracting the air by means of a pump.

"The second method was by employing a pump, or pumps, to extract both the air and the water, which would be applicable in all places, and essential in those cases where there was no well or pit.

"This latter method was the one I then preferred, and is the only one I afterwards continued to use. In Newcomen's engine, the piston is kept tight by water, which could not be applicable in this new method; as, if any of it entered into a partially exhausted and hot cylinder, it would boil and prevent the production of a vacuum, and would also cool the cylinder, by its evaporation during the descent of the piston.

"I proposed to remedy this defect by employing wax, tallow, or other grease, to lubricate and keep the piston tight. It next occurred to me, that the mouth of the cylinder being open, the air which entered to act on the piston would cool the cylinder, and condense some steam on again filling it, I therefore proposed to put an air-tight cover upon the cylinder, with a hole and stuffing-box for the piston-rod to slide through and to admit steam above the piston to act upon it instead of the atmosphere. [N.B. The piston-rod sliding through a stuffing-box was new in steam-engines; it was not necessary in Newcomen's engine, as the mouth of the cylinder was open, and the piston stem was square and very clumsy. The fitting the piston-rod to the piston by a cone was an after improvement of mine (about 1774).] There still remained another source of the destruction of steam, the cooling of the cylinder by the external air, which would produce an internal condensation whenever steam entered it, and which would be repeated every stroke; this I proposed to remedy by an external cylinder containing steam, surrounded by another of wood, or of some other substance which would conduct heat slowly.

"When once the idea of the separate condensation was started, all these improvements followed as corollaries in quick succession, so that in the course of one or two days, the invention was thus far complete in my mind, and I immediately set about an experiment to verify it practically. I took a large brass syringe, 1¾ inches diameter, and 10 inches long, made a cover and bottom to it of tin-plate, with a pipe to convey steam to both ends of the cylinder from the boiler; another pipe to convey steam from the upper end to the condenser (for, to save apparatus, I inverted the cylinder). I drilled a hole longitudinally through the axis of the stem of the piston, and fixed a valve at its lower end, to permit the water which was produced by the condensed steam on first filling the cylinder, to issue. The condenser used upon this occasion consisted of two pipes of thin tin-plate, ten or twelve inches long, and about one-sixth inch diameter, standing perpendicular, and communicating at top with a short horizontal pipe of large diameter, having an aperture on its upper side which was shut by a valve opening upwards. These pipes were joined at bottom to another perpendicular pipe of about an inch diameter, which served for the air and water-pump; and both the condensing pipes and the air-pump were placed in a small cistern filled with cold water. [N.B. This construction of the condenser was employed from knowing that heat penetrated thin plates of metal very quickly, and considering that if no injection was thrown into an exhausted vessel, there would be only the water of which the steam had been composed, and the air which entered with the steam, or through the leaks, to extract.]

"The steam-pipe was adjusted to a small boiler. When steam was produced, it was admitted into the cylinder, and soon issued through the perforation of the rod, and at the valve of the condenser. When it was judged that the air was expelled, the steam-cock was shut, and the air-pump piston-rod was drawn up, which leaving

the small pipes of the condenser in a state of vacuum, the steam entered them and was condensed. The piston of the cylinder immediately rose and lifted a weight of about 18 lbs., which was hung to the lower end of the piston-rod. The exhaustion cock was shut, the steam was readmitted into the cylinder, and the operation was repeated, the quantity of steam consumed, and the weights it could raise were observed, and, excepting the non-application of the steam-case and external covering, the invention was complete, in so far as regarded the savings of steam and fuel.

"A large model, with an outer cylinder and wooden case, was immediately constructed, and the experiments made with it served to verify the expectations I had formed, and to place the advantage of the invention beyond the reach of doubt. It was found convenient afterwards to change the pipe-condenser for an empty vessel, generally of a cylindrical form, into which an injection played, and in consequence of there being more water and air to extract, to enlarge the air-pump.

"The change was made, because, in order to procure a surface sufficiently extensive to condense the steam of a large engine, the pipe-condenser would require to be very voluminous, and because the bad water with which engines are frequently supplied would crust over the thin plates, and prevent their conveying the heat sufficiently quick. The cylinders were also placed with their mouths upwards, and furnished with a working-beam and other apparatus as was usual in the ancient engines; the inversion of the cylinder, or rather of the piston-rod, in the model, being only an expedient to try more easily the new invention, and being subject to many objections in large engines."

13. Development of Watt's Engine: the rotative type. In a second patent (1781) Watt describes the "sun-and-planet" wheels and other methods of making the engine give continuous revolving motion to a shaft provided with a fly-wheel. He had intended to use the crank and connecting-rod, for this purpose (a mechanical device familiar even at that time from its use in the common foot-lathe), and had even made a model of it, but the application of the crank to the steam-engine had meanwhile been patented by one Pickard, and Watt, rather than make terms with Pickard, made use of his sun-and-planet motion until the patent for the application of the crank expired. The reciprocating motion of earlier forms had served only for pumping, but by this invention Watt opened up for the steam-engine a thousand other channels of usefulness. The engine was still single-acting; the connecting rod was attached to the far end of the beam, and that carried a counterpoise which served to raise the piston when steam was admitted below it.

14. Further improvements by Watt. In 1782 Watt patented two further improvements of the first importance, both of which he had invented some years before. One was the use of double action, that is to say, the application of steam and vacuum to each side of the piston alternately. The other, which had

been invented as early as 1769, was the use of steam expansively, in other words, the plan (now used in all engines that aim at

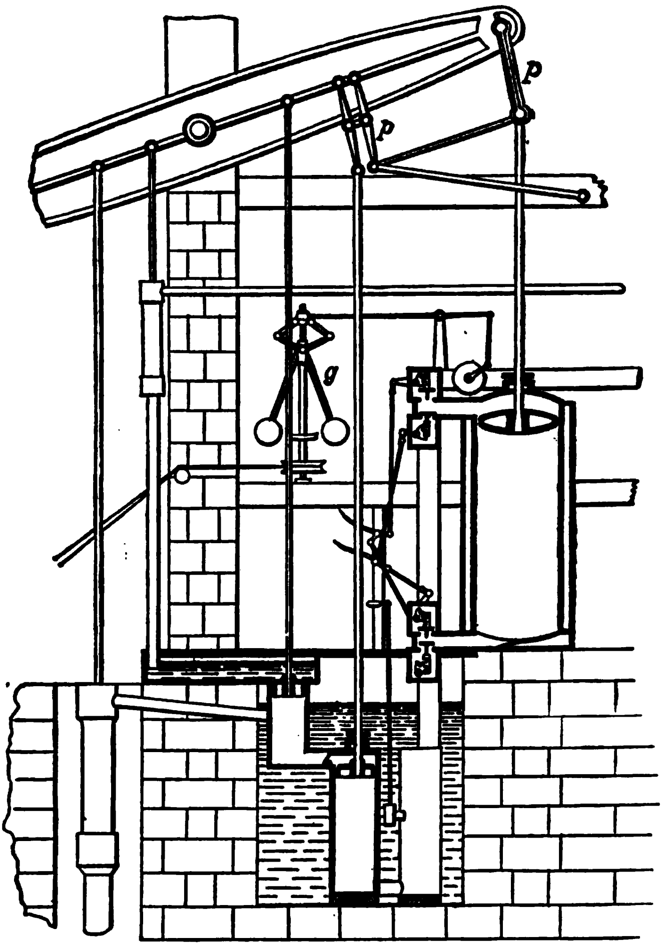


FIG. 7. Watt's Double-acting Engine, 1782.

economy of fuel) of stopping the admission of steam when the piston had made only a part of its stroke, and allowing the rest of the stroke to be performed by the expansion of the steam already in the cylinder. To let the piston push as well as pull the end of the beam Watt devised his so-called parallel motion, an arrangement of links connecting the piston-rod head with the beam in such a way as to guide the rod to move in a very nearly

straight line. He further added the throttle-valve, for regulating the rate of admission of steam, and the centrifugal governor, a double conical pendulum, which controlled the speed by acting on the throttle-valve. The stage of development reached at this time is illustrated by the engine of fig. 7 (from Stuart's *History of the Steam-engine*), which shows the parallel motion *pp*, the governor *g*, the throttle-valve *t*, and a pair of steam and exhaust valves at each end of the cylinder.

Among other inventions of Watt were the "indicator," by which diagrams showing the relation of the steam-pressure in the cylinder to the movement of the piston are automatically drawn; a steam tilt-hammer; and also a steam locomotive for ordinary roads,—but this invention was not prosecuted. As an inventor Watt was skilfully seconded by his assistant Murdoch, to whose ingenuity, he says, are due "many improvements"—amongst them, the introduction of the slide-valve as a means of controlling the admission and release of steam.

In partnership with Matthew Boulton, Watt carried on in Birmingham the manufacture and sale of his engines with the utmost success, and held the field against all rivals in spite of severe assaults on the validity of his patents. A special Act of Parliament was obtained which extended the patent monopoly for a term of twenty-five years from 1775. Notwithstanding Watt's knowledge of the advantage to be gained by using steam expansively he continued to employ only low pressures—seldom more than 7 lbs. per square inch over that of the atmosphere. His boilers were fed, as Newcomen's had been, through an open pipe which rose high enough to let the column of water in it balance the pressure of the steam. Following Savery, he adopted the term "horse-power" as a mode of rating engines and gave it a particular meaning, by defining one horse-power as the rate at which work is done when 33,000 lbs. are raised one foot in one minute. This estimate was based on trials of the work done by horses; it is excessive as a statement of what an average horse can do in working continuously for any long time, but Watt purposely made it excessive in order that his customers might have no reason to complain on this score.

15. Non-condensing Steam-Engines. In the fourth claim in Watt's first patent, the second sentence describes a

non-condensing engine, which would have required steam of a considerably higher pressure than served in the condensing engine. His narrative also shows that he had made experiments in this direction before devising the separate condenser. This, however, was a line of invention which Watt did not follow up, perhaps because so early as 1725 a non-condensing engine had been described by Leupold in his *Theatrum Machinarum*. Leupold's proposed engine (for the main features of which he professes himself indebted to Papin) is shown in fig. 8, which

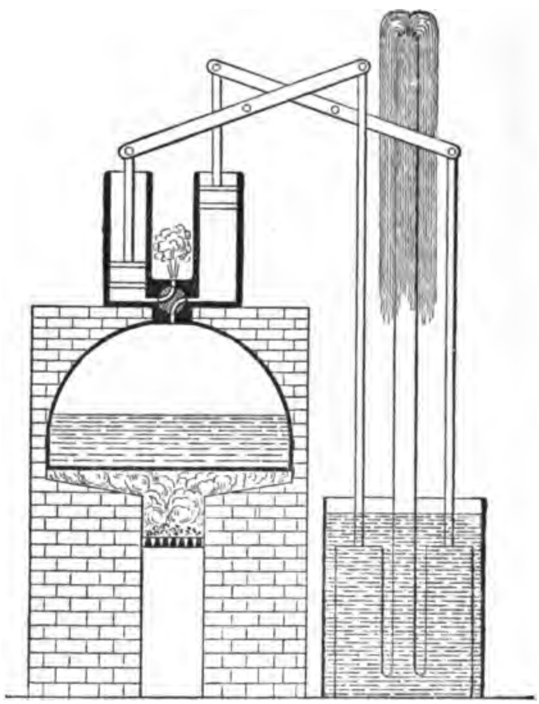


FIG. 8. Non-condensing Engine described by Leupold (1725).

makes its action sufficiently clear. Watt's aversion to high-pressure steam was strong, and its influence on steam-engine practice long survived the expiry of his patents. So much indeed was this the case that the terms "high-pressure" and "non-condensing" were for many years synonymous, in contradistinction to the "low-pressure" or condensing engines of Watt. This nomenclature no longer holds good; in modern practice

many condensing engines use as high pressures as non-condensing engines, and by doing so are able to take advantage of Watt's great invention of expansive working to a degree which was impossible in his own practice.

16. Use of comparatively high-pressure steam. The introduction of the non-condensing and, at that time, relatively high-pressure engine was effected in England by Trevithick and in America by Oliver Evans about 1800. Both Evans and Trevithick applied their engines to propel carriages on roads, and both used for boiler a cylindrical vessel with a cylindrical flue inside—the construction now known as the Cornish boiler. In partnership with Bull, who had been a workman in the employment of Boulton and Watt, Trevithick had previously made direct-acting pumping-engines, with an inverted cylinder set over and in line with the pump-rod, thus dispensing with the beam that had been a feature in all earlier forms. But in these "Bull" engines, as they are called, a condenser was used, or, rather, the steam was condensed by a jet of cold water in the exhaust-pipe, and Boulton and Watt successfully opposed them as infringing the patent for condensation in a separate vessel. To Trevithick belongs the distinguished honour of being the first to use a steam-carriage on a railway; in 1804 he built a locomotive in the modern sense, to run on what had formerly been a horse-tramway in Wales; and it is noteworthy that the exhaust steam was discharged into the funnel to force the furnace draught, a device which, 25 years later, in the hands of George Stephenson, went far to make the locomotive what it is to-day. In this connexion it may be added that as early as 1769 a steam-carriage for roads had been built in France by Cugnot, who used a pair of single-acting high-pressure cylinders to turn a driving axle step by step by means of pawls and ratchet-wheels. To the initiative of Evans may be ascribed the early general use of high-pressure steam in the United States, a feature which for many years distinguished American from English practice.

17. Compound Engines. Hornblower and Woolf. Amongst the contemporaries of Watt one name deserves special mention. In 1781 Jonathan Hornblower constructed and patented what would now be called a compound engine, with two cylinders of different sizes. Steam was first admitted into the smaller

cylinder, and then passed over into the larger, doing work against a piston in each. In Hornblower's engine the two cylinders were placed side by side, and both pistons acted on the same end of a beam overhead. This was an instance of the use of steam expansively, and as such was earlier than the patent, though not earlier than the invention, of expansive working by Watt. Hornblower was crushed by the Birmingham firm for infringing their patent in the use of a separate condenser and air-pump.

Soon after the expiry of Watt's master patent in 1800 the compound engine was revived by Woolf, with whose name it is often associated. Using steam of fairly high pressure, and cutting off the supply before the end of the stroke in the small cylinder, Woolf expanded the steam to six or even nine times its original volume. Mechanically the double-cylinder compound engine has this advantage over an engine in which the same amount of expansion is performed in a single cylinder, that the thrust or pull exerted by the two pistons in the compound engine varies less throughout the action than that which is exerted by the piston of the single-cylinder engine. This advantage may have been clear to Hornblower and Woolf, and to other early users of compound expansion. But another and a more important merit of the system lies in a fact of which neither they nor for many years their followers in the use of compound engines were aware—the fact that by dividing the whole range of expansion into two parts the cylinders in which these are separately performed are subject to a reduced range of fluctuation in their temperature. This, as we shall have occasion to point out more particularly in a later chapter, limits to a great extent a source of waste which is present in all steam-engines, namely, the waste which results from the heating and cooling of the metal by its alternate contact with hot and cooler steam. The system of compound expansion is now used in nearly all large engines that pretend to economy. Its introduction forms the only great improvement which the steam-engine has undergone since the time of Watt; and we are now able to recognize it as a very important step in the direction set forth in his "first principle," that the cylinder should be kept as hot as the steam that enters it.

18. The Cornish Pumping Engine. Woolf introduced the compound engine somewhat widely about 1814, as a pumping

engine in the mines of Cornwall. But it met a strong competitor there in the high-pressure single-cylinder condensing engine, which was at that time developing, in the hands of Trevithick and others, into a machine of great efficiency, and which had an evident advantage over Woolf's in the simplicity of its construction. Woolf's engine fell into comparative disuse, and the single-cylinder type took a form which, under the name of the Cornish pumping engine, was for many years famous for its great economy of fuel. In this engine the cylinder was set under one end of a beam, from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted above the piston for a short portion of the stroke, thereby raising the pump-rod, and was allowed to expand for the remainder. Then an equilibrium valve, connecting the spaces above and below the piston, as in fig. 6, was opened, and the pump-rod descended, doing work in the pump and raising the engine piston. The large mass which had to be started and stopped at each stroke served by its inertia to counterbalance the inequalities of steam pressure which were due to expansive working, for the pump rods and other reciprocating parts stored up energy of motion in the early part of the stroke, when the steam pressure was greatest, and gave out energy in the later part, when expansion had greatly lowered the pressure. The frequency of the stroke was controlled by a device called a cataract, consisting of a small plunger pump, in which the plunger, raised at each stroke by the engine, was allowed to descend more or less slowly by the escape of fluid below it through an adjustable orifice, and in its descent liberated catches which held the steam and exhaust valves from opening. A similar device controlled the equilibrium valve. The cataract could be set to give a pause at the end of the piston's down-stroke, so that the pump cylinder might have time to become completely filled.

The Cornish engine is interesting as the earliest form which achieved an efficiency at all comparable with that of good modern engines. For many years monthly reports were published of the "duty" of these engines, the "duty" being the number of foot-pounds of work done per bushel or (in some cases) per cwt. of coal. The performance of the engines became a matter of almost sporting interest to mining engineers, and no pains were spared to "beat the record." The average duty of engines in the

Cornwall district rose from about 18 millions of foot-pounds per cwt. of coal in 1813 to 68 millions in 1844, after which less effort seems to have been made to maintain a high efficiency[1]. In individual cases much higher results were reported, as in the Fowey Consols engine, which in 1835 was stated to have a duty of 125 millions. This (to use a more modern mode of reckoning) is equivalent to the consumption of only a little more than $1\frac{3}{4}$ lb. of coal per hour per horse-power—a result surpassed by very few engines in even the best recent practice. It is difficult to credit figures which, even in exceptional instances, place the Cornish engine of that period on a level with the most efficient modern engines—in which compound expansion and higher pressure combine to make a much more perfect thermodynamic machine; and apart from this there is room to question the accuracy of the Cornish reports. They played, however, a useful part in the process of steam-engine development by directing attention to the question of efficiency, and by demonstrating the advantage to be gained from high-pressure and expansive working, at a time when the theory of the steam-engine had not yet taken shape.

It may be added that the success of the Cornish type was no doubt largely responsible for the tendency which down to a very recent period engine builders have shown to interpose a beam between the steam-cylinder and the pump or crank on which work is being done. For a long time the beam appears, in one form or another, as an almost inevitable part of a steam-engine. The lesson to be learnt from Bull's early direct-acting engine was apparently, in general, overlooked.

19. **Revival of the Compound Engine.** The final revival of the compound engine did not occur until about the middle of the century, and then several agencies combined to bring it about. In 1845 M'Naught introduced a plan of improving beam engines of the original Watt type, by adding a small high-pressure cylinder with a piston acting on the beam between the centre and the fly-wheel end. Steam of higher pressure than had formerly been used, after doing work in the new cylinder, passed into the old or low-pressure cylinder, where it was further expanded. Many engines whose power was proving insufficient for the extended

[1] *Min. Proc. Inst. C.E.*, vol. xxiii., 1863.

machinery they had to drive were "M'Naughted" in this way, and after conversion were found not only to exert more power but to show a marked economy of fuel. The compound form was selected by Mr Pole for the pumping engines of Lambeth and other waterworks about 1850; in 1854 John Elder began to use it in marine engines; in 1857 Mr E. A. Cowper added a steam-jacketed intermediate reservoir for steam between the high and low-pressure cylinders, which made it unnecessary for the low-pressure piston to be just beginning when the other piston was just ending its stroke. As the mechanical construction of engines and boilers improved and facilities therefore increased for the use of high-pressure steam, compound expansion became more and more general, its advantage becoming more conspicuous with every increase in boiler pressure. Now-a-days there are few large land engines and scarcely any marine engines that are not compound. In marine practice especially, where economy of fuel is a much more important factor in determining the design than it is on land, the principle of compound expansion has been greatly extended by the general introduction of triple and occasionally even of quadruple expansion engines, in which the steam is made to expand successively in three or in four cylinders. Even in locomotives for railways, where other considerations are of more moment than the saving of coal, compound expansion has found a place, though its use there is by no means general.

The growth of compound expansion has been referred to at some length, because it forms the most distinctive improvement which the steam-engine has undergone since the time of Watt. For the rest, the progress of the steam-engine has consisted in its adaptation to particular uses, in the invention of features of mechanical detail, in the recognition and application of thermodynamical principles, in better structural design and in improved methods of manufacture by which it has profited in common with all other machines. These have made possible the use of steam with eight or ten times the pressure of that employed by Watt, and have allowed the mean speed of movement of the piston to be greatly increased, with consequent gains both in the amount of power obtainable from an engine of given size and in the efficiency of the action.

20. Application to locomotives. The adaptation of the

steam-engine to railways, begun by Trevithick, became a success in the hands of George Stephenson, whose engine the "Rocket," when tried along with others on the Stockton and Darlington railroad in 1829, not only distanced its competitors but settled once and for all the question whether horse traction or steam traction was to be used on railways. The principal features of the "Rocket" were an improved steam-blast for urging the combustion of coal and a boiler (suggested by Booth, the secretary of the railway) in which a large heating surface was given by the use of many small tubes through which the hot gases passed. Further, the cylinders, instead of being vertical as in earlier locomotives, were set at a slope, which was afterwards altered to a position still more nearly horizontal. To these features there was added later the "link motion," a contrivance which enabled the engine to be quickly reversed and the amount of expansion to be readily varied. In the hands of George Stephenson and his son Robert the locomotive took a form which has been in all essentials maintained by the far heavier locomotives of modern practice.

21. **Application to steamboats.** The first practical steamboat was the tug "Charlotte Dundas," built by William Symington, and tried in the Forth and Clyde Canal in 1802. A Watt double-acting condensing engine, placed horizontally, acted directly by a connecting-rod on the crank of a shaft at the stern, which carried a revolving paddle-wheel. The trial was successful, but steam towing was abandoned for fear of injuring the banks of the canal. Ten years later Henry Bell built the "Comet," with side paddle-wheels, which ran as a passenger steamer on the Clyde; but an earlier inventor to follow up Symington's success was the American Robert Fulton, who, after unsuccessful experiments on the Seine, fitted a steamer on the Hudson in 1807 with engines made to his designs by Boulton and Watt, and brought steam navigation for the first time to commercial success.

The American river boats soon began to use high-pressure steam, but English engineers looked askance on a practice which led to frequent explosions. They were moreover slow to realise that high pressure is a necessary condition of economical working. In 1835 it was usual for the pressure in marine boilers to be no more than 4 or 5 pounds per square inch above the pressure of the atmosphere, and for many years later pressures of 20 or 25

pounds were common. With the introduction of compound working and with the substitution of cylindrical boilers for the weak box-boiler originally used on board ship the pressure rose considerably. In 1872 Sir F. Bramwell, describing the typical marine practice of that time[1], gives a list of engines—all compound—in which the pressure ranged from 45 to 60 pounds. The consumption of coal in these engines was generally from 2 to 2½ pounds per hour per indicated horse-power, and the mean piston speed was about 350 feet per minute. Nine years later Mr F. C. Marshall gives a similar list[2], in which the mean pressure is 77 pounds, the mean piston speed about 460 feet per minute, and the consumption of coal a trifle under 2 pounds per hour per indicated horse-power. These engines were also of the type in which steam is successively expanded in two cylinders. The triple expansion type of engine came into general use shortly after that date and led at once to a marked advance in boiler pressure with a considerable gain in economy of fuel. Reviewing the progress of marine engineering in the decade from 1881 to 1891 Mr Blechynden[3] gives a list of triple engines with boiler pressures of about 160 pounds and piston speeds of about 500 or 600 feet per minute. These engines consume on the average about 1½ pounds of coal per indicated horse-power-hour[4].

The progressive rise in steam-pressure and in piston speed has not only increased the efficiency of engines but has greatly reduced their bulk for a given power. The rate at which work is done per square inch of piston area is equal to the mean piston speed multiplied by the mean effective pressure, and increased pressure of admission implies increased mean pressure throughout the stroke.

The remarkable efficiency now reached by the marine engine is in part due to its great size; a big engine being, *cæteris paribus*, rather more efficient than a small one. The rapid growth in size is a marked feature of recent progress. Ten thousand indicated horse-power as the performance of a single set of engines is not unusual in a first-rate Atlantic liner or in a war vessel. The

[1] *Proc. Inst. Mech. Eng.*, 1872.

[2] *Proc. Inst. Mech. Eng.*, 1881.

[3] *Proc. Inst. Mech. Eng.*, 1891.

[4] On this subject see further a paper by Sir A. J. Durston on the progress of Marine Engineering, read at the International Congress of Naval Architects, 1897. *Engineering*, July 9—16, 1897.

twin-screw engines of the "Paris" or of the "New York" develope together some twenty thousand horse-power: those of the "Campania" and "Lucania" (1893) work at 31,000 horse-power, with a boiler pressure of 165 pounds per square inch and a mean piston speed of about 1000 feet per minute. We shall have occasion in later chapters to mention some of the forms which the steam-engine has assumed in present day practice, and to state more particularly the results which have been found in trials made to determine the efficiency.

22. Development of the Theory of Heat-Engines. It is remarkable how little the infancy of the steam-engine has owed to scientific nursing. The early inventors had no theory of thermodynamics to guide them. Watt had the advantage, as he mentions in his narrative, of a knowledge of Black's doctrine of latent heat; but there was no philosophy of the relation of work to heat until long after the inventions of Watt were complete. The theory of the steam-engine as a heat-engine may be said to date from 1824, when Carnot published his *Réflexions sur la Puissance Motrice du Feu.* He there showed that heat does work only by being let down from a higher to a lower temperature. But Carnot had no idea that any of the heat disappears in the process, and it was not until the doctrine of the conservation of energy was established in 1843 by the experiments of Joule, which determined the mechanical equivalent of heat, that the theory of heat-engines began a vigorous growth. Important data were furnished by Regnault's classical experiments on the properties of steam, the results of which were published in 1847. From 1849 onwards the science of thermodynamics was developed with extraordinary rapidity by Clausius, Rankine, and Thomson (Lord Kelvin), and was applied, especially by Rankine, to practical problems in the use of steam. The publication in 1859 of Rankine's *Manual of the Steam-Engine* formed an epoch in the philosophical treatment of the subject and gave steam engineers the opportunity of ceasing to be mere empirics. Unfortunately, however, for its bearing on practice, while the thermodynamic theory was rigorous in itself, the application of it to steam-engine problems was to a great extent founded on certain simplifying assumptions which experience has since shown to be far from correct. It was assumed that the cylinder and piston might be treated as

behaving to the steam like non-conducting bodies,—that the transfer of heat between the steam and the metal was negligibly small. Rankine's calculations of steam-consumption, of work, and of thermodynamic efficiency involve this assumption, except in the case of steam-jacketed cylinders, where he estimates that the steam in its passage through the cylinder takes just enough heat from the jacket to prevent a small amount of condensation which would otherwise occur as the process of expansion goes on. These assumptions are not supported by experiment. If the transfer of heat from steam to metal could be overlooked, the steam which enters the cylinder would remain during admission as dry as it was before it entered, and the volume of steam consumed per stroke would correspond with the volume of the cylinder up to the point of cut-off. It is here that the actual behaviour of steam in the cylinder diverges most widely from the behaviour which has been assumed. When steam enters the cylinder it finds the metal chilled by the previous exhaust, and a portion of it is at once condensed. This has the effect of increasing, often very largely, the volume of boiler steam required per stroke. As expansion goes on the water that was condensed during admission begins to be re-evaporated from the sides of the cylinder, and this action is generally continued during the escape of the steam. In later chapters the effect which this exchange of heat between the metal of the cylinder and the working fluid produces on the economy of the engine will be discussed, and an account will be given of experimental means by which we may examine the amount of steam that is initially condensed and trace its subsequent re-evaporation. The influence which the walls of the cylinder exert is in fact immense, by the alternate give and take of heat between them and the steam. It is now recognized that any theory which fails to take account of these exchanges of heat fails also to yield even comparatively correct results in calculating the relative efficiency of various steam pressures or various ranges of expansion. But the exchanges of heat are so complex that there seems little prospect of submitting them to any comprehensive theoretical treatment, and we must rather look for help in the future development of engines to the scientific analysis of experiments made upon actual machines. Many such experiments have been made and their value is now fully realised, by no persons more than by the designers of the best modern engines.

Questions relating to the influence on thermal economy of speed, of pressure, of ratio of expansion, of jacketing, of compound expansion, or of superheating must in the main be settled by an appeal to experiment. The student must not, however, conclude that because the conditions under which an actual engine works are so complex as to make an exact theory of the action impracticable, no theory need be studied. The very complexity of conditions makes the study of theory more necessary, as a guide in judging what conditions are favourable to efficiency and what are unfavourable. Moreover the general theory of heat-engines gives the steam engineer a counsel of perfection, by assigning a limit of efficiency which engines may approach but cannot surpass. Even to interpret rightly the results of experiments requires a knowledge both of the principles of thermodynamics and of the physical properties of steam.

References.—Dircks, *Life of the Marquis of Worcester*, 1865, containing a reprint of the *Century of Inventions* (1663). Desaguliers, *Course of Experimental Philosophy*, 1763. Robison, *System of Mechanical Philosophy*, Vol. II. 1822. Stuart, *Descriptive History of the Steam-Engine*, 1825; Farey, *Treatise on the Steam-Engine*, 1827; Tredgold, *The Steam-Engine*, 1838; Muirhead, *Mechanical Inventions of James Watt*, and *Life of Watt*; Galloway, *The Steam-Engine and its Inventors*; Thurston, *History of the Growth of the Steam-Engine*; Cowper on the Steam-Engine (*Heat Lectures, Inst. C.E.*, 1884).

CHAPTER II.

ELEMENTARY THEORY OF HEAT-ENGINES.

23. Laws of Thermodynamics. The First Law. In the action of a heat-engine, heat is either taken in by the engine from a furnace or from some external source or is generated by the combustion of fuel within the engine itself. A portion of the heat thus supplied is spent in doing mechanical work and so ceases to exist as heat, being converted into another form of energy; and the remainder is rejected by the engine, still in the form of heat. The relation which holds between the heat supplied, the heat converted into mechanical energy, and the heat rejected depends on two general principles which are described as the two Laws of Thermodynamics. The first law states the fact that the amount of heat which disappears in the process (as heat) is proportional to the amount of mechanical work done in the engine; in other words, it states the principle of the Conservation of Energy in relation to the doing of mechanical work by the agency of heat. This may be expressed in the following terms:—*When mechanical energy is produced from heat a definite quantity of heat goes out of existence for every unit of work done; and conversely, when heat is produced by the expenditure of mechanical energy the same definite quantity of heat comes into existence for every unit of work spent.*

To put this statement into a numerical form we must have a unit for the measurement of quantities of heat as well as a unit for the measurement of mechanical work. For engineering purposes the foot-pound is the common unit of work. This convenient and familiar unit is open to the objection that it has slightly different values in different places on account of differences in the intensity

of gravity; but these differences are scarcely large enough to be important from a practical point of view. In cases where greater precision of statement is required a particular locality or rather a particular latitude has to be specified, or recourse may be had to absolute units, such as the foot-poundal or the erg, which are independent of gravity.

Quantities of heat are expressed in terms of the *thermal unit*, which is the quantity of heat required to raise the temperature of 1 lb. of water by 1 degree. Fahrenheit's scale is generally used by English engineers in stating temperature, and so the Fahrenheit degree is to be understood in this definition. The corresponding unit of heat on the Centigrade mode of reckoning would be greater in the proportion of nine to five. To make the definition of the thermal unit precise we have to specify at what place in the scale of temperature the change through one degree is supposed to occur, for the specific heat of water is not quite constant. According to Regnault, it takes rather more heat to raise the temperature of a pound of water 1 degree if the temperature is high than if it is low. Later experiments, however, show that when water is warmed from the temperature of melting ice its specific heat at first decreases slightly as the temperature rises, though at higher temperatures it increases. By Rankine and others the standard temperature assumed in the definition is that of the maximum density of water, or about 39° Fah.: later writers have preferred to take a standard temperature of about 60° Fah. Further, for the purpose of exact definition it is necessary not only to specify the standard temperature, but also to say whether the 1 degree interval of temperature is to be taken on the scale of the mercury thermometer or on that of the air or other gas thermometer.

Our knowledge of the mechanical equivalent of heat is originally due to the experiments of Joule, which were begun in 1843 and continued for many years. Causing the potential energy of a raised weight to be spent in turning a paddle which generated heat by the agitation of the liquid in which it was immersed and observing the increase in temperature which this brought about, Joule arrived at the figure 772 as the number of foot-pounds equivalent to one thermal (Fahrenheit) unit, and this was for long the commonly accepted value of the mechanical equivalent of heat. Later experiments by Joule himself gave a larger number; in

1878 an improved method of measurement, in which the mechanical stirring of water was still used, pointed to a value between 774 and 775. A comparison by Rowland[1] of the scale of the thermometer used by Joule with that of an air thermometer led to a further increase in this number, bringing it to about 778, the standard temperature being about 60° Fah. and the interval of 1 degree being taken on the air thermometer. This value has received confirmation from the experiments of Rowland himself and more recently from those of Griffiths[2], which were conducted by entirely different methods. The results of Griffiths and other recent investigators point indeed to a somewhat higher value still. An important determination by Osborne Reynolds and W. M. Moorby[3], of the amount of work spent in raising the temperature of water from 32° Fah. to 212° Fah. gives very approximately 778 foot-pounds as the mean value of the mechanical equivalent throughout that range: in other words, 180 times 778 is the number of foot-pounds of work required to raise the temperature of 1 lb. of water from 32° to 212°. This mode of defining and measuring the mechanical equivalent has the advantage of escaping all ambiguity in regard to the thermometric scale used in specifying the unit of heat. Taking the evidence together there can be no doubt that the old number is too low, and that the mechanical equivalent of one thermal unit is at least 778 and perhaps as much as 779 or even 780 foot-pounds. The number 778 is used in the calculations that occur in this book. Since a definite number of foot-pounds is equivalent to 1 thermal unit, we may, if we please, express quantities of work in thermal units, or quantities of heat in foot-pounds[4].

[1] *Proceedings of the American Academy*, 1879.

[2] *Phil. Trans. Roy. Soc.* 1893.

[3] *Phil. Trans.* 1897.

[4] To escape the uncertainty which attaches to a definition of the unit of heat based on the heating of water, suggestions have been made to adopt as unit the amount of heat which is required to melt unit mass of ice, or the amount which is required to evaporate unit mass of water, under a specified pressure, but such units have not come into use. A committee of the British Association reporting in 1896 on a thermal unit have recommended the use of a dynamical unit of heat, namely $4{\cdot}2 \times 10^7$ ergs. This number is (according to the most recent experiments) approximately equivalent to the heat required to raise 1 gramme of water through 1° C. (on the gas thermometer) at ordinary temperatures; and it may be taken as representing the heat required to do this at some standard temperature, which is not as yet defined. This definition amounts to a statement that $4{\cdot}2 \times 10^7$ ergs are a mechanical equivalent of 1 gramme-degree, leaving the standard

24. The Second Law of Thermodynamics. *It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.*

This is the form in which the second law has been stated by Clausius[1]. Another statement of it, different in form but similar in effect, has been given by Lord Kelvin[2]. Its force may not be immediately obvious, but it will be shown below that this law sets a most important limit to the convertibility of heat into work. So far as the first law goes, there is nothing to prevent the whole heat taken in by an engine from changing into mechanical energy. In consequence of the second law, however, as we shall presently see, no heat-engine converts, or can convert, more than a small fraction of the heat supplied to it into work; a large part is necessarily rejected as heat. The ratio

$$\frac{\text{Heat converted into work}}{\text{Heat taken in by the engine}}$$

is a fraction always much less than unity. This fraction is called the *efficiency* of the engine considered as a heat-engine.

25. The Working Substance in a Heat-Engine. In every heat-engine there is a *working substance* which alternately takes in and rejects heat. In general it suffers changes of volume, and does work by overcoming resistance to these changes. The working substance may be gaseous, liquid, or solid. We can, for example, imagine a heat-engine in which the working substance is a long metallic rod, arranged to act as the pawl of a ratchet-wheel with closely pitched teeth. Let the rod be heated so that

temperature unspecified until further knowledge of the specific heat of water is obtained. The name *Calory* is proposed for this unit. Since 1 foot-pound (in the latitude of Greenwich) is $1{\cdot}3565 \times 10^7$ ergs, this calory is, according to this definition, the heat-equivalent of $\frac{4{\cdot}2}{1{\cdot}3565}$ or 3·096 foot-pounds. Further, as there are 453·6 grammes in 1 lb., and 1·8° C. in 1° Fah., the British thermal unit is equal to $\frac{453{\cdot}6}{1{\cdot}8}$ or 252 gramme-degrees. Hence if the calory as above defined be taken as representing one gramme degree, the mechanical equivalent of the British thermal unit would be $252 \times 3{\cdot}096$ or 780 foot-pounds.

A useful summary of results obtained by various observers will be found in a paper by Mr Griffiths on the thermal unit, *Phil. Mag.* Nov. 1895.

[1] See Clausius, *Mechanical Theory of Heat*, translated by W. R. Browne.

[2] See Lord Kelvin's (Sir W. Thomson's) *Collected Papers*, Vol. I., for his early investigations in thermodynamic theory.

it elongates sufficiently to drive the wheel forward through the space of one tooth. Then let the rod be cooled (say by applying cold water), the ratchet-wheel being meanwhile held from returning by a separate click or detent. The rod, on cooling, will retract so as to engage itself with the next succeeding tooth, which may then be driven forward by heating the rod again, and so on. To make it evident that such an engine would do work, we have only to suppose that the ratchet-wheel carries round with it a drum by which a weight is wound up. The device forms a complete heat-engine, in which the working substance is a solid rod, which receives heat by being brought into contact with some source of heat at a comparatively high temperature, transforms a small part of this heat into work, and rejects the remainder to what we may call a receiver of heat, which is kept at a comparatively low temperature. The greater part of the heat may be said simply to pass through the engine, from the source to the receiver, *becoming degraded as regards temperature* as it goes. It will be seen presently that this is typical of the action of all heat-engines; when they are doing work they must take in heat at a comparatively high temperature and reject heat at a comparatively low temperature. They convert some heat into work only by letting down a much larger quantity of heat from a high to a relatively low temperature. The action is, to some extent, analogous to that of a water-wheel, which does work by letting down water from a high to a lower level, change of level in the one case being the analogue of change of temperature in the other. But there is this important difference, that whereas in the action of the water-wheel none of the water disappears, in the action of the heat-engine an amount of heat disappears which is equivalent to the work done.

26. Graphic Representation of Work done in the changes of volume of a Fluid. In almost all actual heat-engines the working substance is a fluid. In some it is air, in some a mixture of several gases. In the steam-engine the working fluid is a mixture (in varying proportions) of water and steam. With a fluid for working substance, work is done by changes óf volume only; its amount depends solely on the relation of pressure to volume during the change, and not at all on the form of the vessels in which the change takes place. Let a diagram be drawn (fig. 9) in which the relation of the intensity of pressure to the volume of

any supposed working fluid is graphically exhibited by the line ABC, where AM, CN are pressures and AP, CQ are volumes, then the work done by the substance in expanding from volume AP to volume CQ is the area of the figure $MABCN$. And similarly, if the substance be compressed from volume CQ back to its original volume in such a manner that the line CDA represents the relation of pressure and volume during compression, a quantity of work is done *upon* the substance which is represented by the area $NCDAM$. Taking the two operations together, we find that the

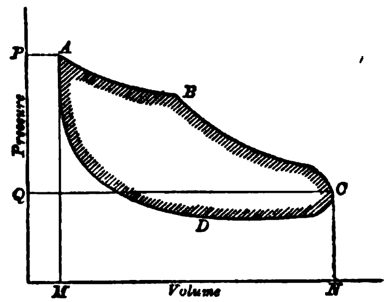


FIG. 9.

substance has done a net amount of work equal to the area of the shaded figure $ABCDA$, or $\int PdV$. This is an example and a generalization of the method of representing work which Watt introduced by his invention of the indicator; the figure $ABCDA$ may be called the *indicator diagram* of the supposed action.

27. Cycle of operations of the working substance. Modern forms of the indicator will be described in a later chapter. For the present it may suffice to say that the indicator draws automatically a diagram showing the relation of the pressure of the working fluid to the movement of the piston, or in other words to the volume of working fluid in the cylinder, and thus gives complete information as to the work done throughout the stroke. Generally in heat-engines the working substance returns periodically to the same state of temperature, pressure, volume, and physical condition. Each time this has occurred the substance is said to have passed through a complete cycle of operations. For example, in a condensing steam-engine, water taken from the hot-well is pumped into the boiler; it then passes into the cylinder as steam, passes thence into the condenser, and thence again as water

into the hot-well; it completes the cycle by returning to the same condition as at first. In other less obvious cases, as in that of the non-condensing steam-engine, a little consideration will show that the cycle is completed, not indeed by the same portion of working substance being returned to the boiler, but by an equal quantity of water being fed to it, while the steam which has been discharged into the atmosphere cools to the temperature of the feed-water. In the theory of heat-engines it is of the first importance to consider as a whole the cycle of operations performed by the working substance (as was first done by Carnot in 1824). If we stop short of the completion of a cycle matters are complicated by the fact that the substance is in a state different from its initial state, and may therefore have changed its stock of internal energy. After the cycle is completed, on the other hand, the internal energy of the substance is necessarily the same as at first, since the condition is in every respect the same. Hence in regard to the cyclic process as a whole this equation must hold good,

$$\text{Heat taken in} = \text{work done} + \text{heat rejected.}$$

28. Engine using a perfect gas as working substance. It is convenient to approach the theory of heat-engines by considering, in the first instance, the action of an engine in which the working substance is any one of the so-called permanent gases, or a mixture of them, such as air. The word permanent, as applied to a gas, is to be understood only as meaning that the gas is liquefied with difficulty—by the use of extremely low temperature in conjunction, generally, with high pressure. So long as gases are under conditions of pressure and temperature widely different from those which produce liquefaction, they conform very approximately to certain simple laws—laws which may be regarded as *rigorously* applicable to ideal substances called *perfect* gases. After stating these laws we shall examine the efficiency of a heat-engine using a gas in a certain manner as working substance, and then show that the results so derived have a general application to all heat-engines whatsoever. In this procedure there is no sacrifice of generality, and a part of the process is of independent service in the discussion of actual air-engines.

The laws which have now to be stated are very nearly though not absolutely true for air, oxygen, nitrogen, hydrogen and carbonic oxide, except when at specially high pressures or specially

low temperatures. Hydrogen probably comes nearest to the ideal of a perfect gas; but no real gas is in this sense strictly perfect.

29. Laws of the permanent gases. Boyle's law. The laws which are very approximately true of the permanent gases, and may be regarded as strictly true of the ideal perfect gas, are the following:—

LAW 1 (Boyle). *The volume of a given mass of gas varies inversely as the pressure, provided the temperature be kept constant.*

Thus, if V be the volume of a given quantity of any gas, and P the pressure, then so long as the temperature is unchanged—

V varies inversely as P, or $PV = \text{constant}$.

For air the value of this constant is 26220 when the temperature is 32° F., V being taken as the volume in cubic feet of 1 lb. of air, and P being expressed in pounds per square foot.

30. Charles's law. LAW 2 (Charles). *Under constant pressure equal volumes of different gases increase equally for the same increment of temperature. Also, if a gas be heated under constant pressure, equal increments of its volume correspond very nearly to equal intervals of temperature as determined by the scale of a mercury thermometer.*

If, for example, we take a vessel containing a quantity of air and heat it from one temperature to another, taking care to arrange the experiment so that the air may expand without any change in its pressure, we shall find that a certain change of volume takes place. Let any other permanent gas then be substituted for the air in the vessel and let the experiment be repeated by heating this other gas from the same initial to the same final temperature as before, the pressure being still kept constant. The volume will be found to have changed by sensibly the same amount as was observed in the experiment with air. And further, if the experiment be varied by using a greater or smaller interval of temperature, it will be found that the change of volume undergone by the air or by any other gas that may be substituted for it is very approximately proportional to the magnitude of the interval of temperature as measured on the scale of the ordinary mercury thermometer. This is equivalent to saying that if we use an air-thermometer (where the air is allowed to

expand without change of pressure) to measure temperatures, defining equal intervals of temperature to be those which correspond to equal expansions on the part of the air, we obtain a thermometric scale which is in substantial though not perfect agreement with the usual mercurial scale, which defines equal intervals of temperature to be those that correspond to equal expansions of mercury in glass.

Experiment shows that the amount by which a gas expands when its temperature is changed by one degree Fahrenheit, the pressure being kept constant, is about $\frac{1}{493}$ of its volume at 32° F. Thus if we take 493 cubic inches of air or any other permanent gas at the temperature 32° and heat it to 33° its volume alters to 494 cubic inches. If we heat it to 34° its volume becomes 495 cubic inches and so on. Similarly if the gas be cooled from 32° F. to 31° F. its volume changes from the original 493 cubic inches to 492, and so on.

Putting this in a tabular form, let the volume be

	493 at 32° F.
It will become	492 at 31° F.
	⋮ ⋮
	461 at 0° F.
	⋮
and finally would be	0 at − 461° F.,

if the same law could be held to apply at indefinitely low temperatures. Any actual gas would change its physical state before so low a temperature were reached.

31. Absolute temperature. The above result may be concisely expressed by saying that if temperature be reckoned, not from the ordinary zero but from a point 461° below the zero of Fahrenheit's scale, the volume of a given quantity of a gas, kept at constant pressure, is proportional to the temperature reckoned from that zero. Temperatures so reckoned are called absolute temperatures, and the point − 461° F. is called the absolute zero of temperature. Denoting any temperature according to the ordinary scale by t, and the corresponding absolute temperature by τ, we have

$$\tau = t + 461 \text{ on the Fahrenheit scale,}$$

and

$$\tau = t + 274 \text{ on the Centigrade scale}^1.$$

[1] The position here assigned to the absolute zero of temperature is that given by Rankine (*Steam-Engine*, p. 226) and is based on a result quoted from Regnault

Charles's law shows that if temperatures be measured by thermometers in which the expanding substance is air, hydrogen, oxygen, or any other permanent gas, those intervals of temperature being called equal which correspond to equal amounts of expansion, then the indications of these thermometers always agree very closely with each other, and also agree, though less closely, with the indications of a mercury thermometer. It will be shown later that the theory of heat-engines affords a means of forming a truly absolute scale of temperature, in the sense that it is a scale which is independent of the properties, as to expansion, of any substance.

It will be further seen that this scale has the same absolute zero as we have arrived at here by considering the properties of the permanent gases, and also makes those intervals equal which are reckoned to be equal on the scale of the gas thermometer.

We are therefore justified in the use of the term absolute, as applied to temperatures measured by the expansion of a gas.

32. Connection between Pressure, Volume, and Temperature in a gas. By Boyle's law we have $P \propto \frac{1}{V}$ where the temperature is kept constant, V being the volume of a given quantity of any gas. By Charles's law we have $P \propto \tau$ when V is kept constant, τ being the absolute temperature. Combining the two laws, we have, for a given mass of any gas,

$$PV = c\tau, \qquad \text{..............................(1)}$$

where c is a constant depending on the specific density of the gas and on the units in which P and V are measured. In what follows it will be assumed that P is measured in pounds per square foot, that V is the volume of 1 lb. in cubic feet, and that τ is the absolute temperature expressed in Fahrenheit degrees. For air, with these units,

$$PV = 53{\cdot}18\tau.$$

that the expansion of a gas from 32° to 212° Fah. is 0·365 of original volume at 32°. The figure in question relates to the expansion of air in a rarefied state, under a pressure about one-fifth that of the atmosphere. In air at atmospheric pressure, however, it appears that Regnault's most authoritative results give an expansion of 0·3665. This would correspond to $\frac{1}{491}$ of the volume at 32° Fah., per 1° Fah. and $\frac{1}{273}$ of the volume at 0° C. per 1° C. Hence the absolute zero is often given in text-books as $-273°$ C. or $-459°$ Fah. The precise position of the absolute zero is uncertain, and the author has not thought it necessary to depart from the number given by Rankine, especially as the ideal condition of a perfect gas is more closely reached in experiments made under low pressures.

33. The Specific Heat of a gas. LAW 3 (Regnault). *The specific heat at constant pressure is constant for any gas.*

By specific heat at constant pressure is meant the heat taken in by 1 lb. of a substance when its temperature rises 1° F., while the pressure remains unchanged—the volume being allowed to change. The law states that this quantity is the same for any one gas, no matter what be the temperature, or what the constant pressure, at which the process of heating takes place.

Another important quantity in the theory of heat-engines is the specific heat at constant volume, that is, the heat taken in by 1 lb. of the substance when its temperature rises 1° F. while the volume remains unchanged—the pressure being free to change. We shall denote specific heat at constant pressure by K_p and specific heat at constant volume by K_v. An obvious difference between the heating of a gas at constant pressure and at constant volume is that when heated at constant volume the gas does no work, whereas heating at constant pressure involves expansion of the gas and consequently the doing of an amount of work equal to the product of the pressure and the increase of volume. Let 1 lb. of a gas be heated at constant pressure P from temperature τ_1 to temperature τ_2 (absolute). Let V_1 be the volume at τ_1 and V_2 the volume at τ_2. Heat is taken in, and external work is done by the expansion of the gas, namely—

$$\text{Heat taken in} = K_p(\tau_2 - \tau_1).$$

$$\text{Work done} = P(V_2 - V_1) = c(\tau_2 - \tau_1).$$

The difference between these quantities, or $(K_p - c)(\tau_2 - \tau_1)$, is the amount by which the stock of internal energy possessed by the gas has increased during the process. It will be shown immediately that this gain of internal energy is the same when the gas has its temperature changed in any other manner from τ_1 to τ_2 and is independent of the condition of the gas as to pressure.

34. The Internal Energy of a gas. LAW 4 (Joule). *When a gas expands without doing external work, and without taking in or giving out heat (and therefore without changing its stock of internal energy), its temperature does not change.*

This fact was established by the experiments of Joule. He connected a vessel containing compressed gas with another

vessel which was empty, by means of a pipe with a closed stop-cock. Both vessels were immersed in a tub of water and were allowed to assume a uniform temperature. Then the stop-cock was opened, and the gas distributed itself between the two vessels, expanding without doing external work. After this the temperature of the water in the tub was found to have undergone no change. The temperature of the gas was unaltered, and no heat had been taken in or given out by it, and no work had been done by it.

Since the gas had neither gained nor lost heat, and had done no work, its internal energy was the same at the end as at the beginning of the experiment. The pressure and volume had changed, but the temperature had not. The conclusion follows that the internal energy of a given quantity of a gas depends only on its temperature, and not upon its pressure or volume; in other words, a change of pressure and volume not associated with a change of temperature does not alter the internal energy. Hence in any change of temperature the change of internal energy is independent of the relation of pressure to volume during the operation: it depends only on the amount by which the temperature has been changed.

Later experiments by Joule and Lord Kelvin have shown that in air and other real gases there is a small fall of temperature when the gas expands without doing work[1]. In other words, there is an appreciable deviation from Joule's Law, which like the other laws stated here is to be regarded as strictly true only in the case of an ideal perfect gas.

To express the quantity of energy which becomes stored up in a gas when its temperature rises, or is extracted from the gas when its temperature falls, we may consider either the case of heating at constant volume, or at constant pressure, since the internal energy depends on the temperature and on nothing else.

In the operation of heating any substance we have

Heat taken in = work done + increase of internal energy.

Take the case of heating at constant volume, and suppose a

[1] See Lord Kelvin's *Collected Papers*, Vol. I. p. 333. The fall of temperature, small as it is, which occurs when a compressed gas is allowed to escape through a constricted orifice forms the basis of Linde's regenerative process of obtaining excessively low temperatures, which he has successfully applied to the liquefaction of air and to the separation of the oxygen of air from the nitrogen.

lb. of gas (assumed perfect or sensibly perfect) to be so heated from absolute temperature τ_1 to absolute temperature τ_2. The heat taken in is

$$K_v(\tau_2-\tau_1)$$

by definition of K_v, the specific heat at constant volume. No external work is done, and hence the whole of this heat goes to increase the stock of internal energy. But in whatever way the temperature be changed from τ_1 to τ_2 the change of internal energy is the same. Hence this expression

$$K_v(\tau_2-\tau_1)$$

measures the change of internal energy which 1 lb. of the gas suffers whenever its temperature changes from τ_1 to τ_2 in any manner whatsoever, no matter how the volume and the pressure vary during the process.

35. Relation between the two Specific Heats. We are now in a position to establish a relation between the two specific heats of a gas, K_v and K_p. It was seen by § 33 that when a gas is heated from τ_1 to τ_2 in one particular way, namely, at constant pressure, the change of its internal energy per lb. may be expressed as

$$(K_p-c)(\tau_2-\tau_1).$$

This expression must agree with the one just found, and hence

$$K_v=K_p-c \quad\text{......................}(2).$$

The ratio $\dfrac{K_p}{K_v}$ enters into many thermodynamic equations and is usually denoted by the letter γ. Using this symbol the above equation may be written

$$K_v=\frac{c}{\gamma-1} \quad\text{.........................}(3).$$

36. Values of the constants for Air. The constant 26220 given in § 29 as the value of PV when V is the volume in cubic feet of 1 lb. of dry air at 32° F., and P is the pressure in pounds per square foot, is derived from a measurement of the density of air by Regnault. Taking the absolute zero of temperature to be 461 degrees below the zero of Fahrenheit's scale, we divide this number by 493 to find c in the formula $PV=c\tau$. This makes $c=53{\cdot}18$.

Regnault's measurements of the specific heat of dry air give 0·2375 thermal units as the value of K_p. Taking J to be 778, this is equivalent to 184·8 foot-pounds. Subtracting c from this we find K_v to be 131·6 foot-pounds, or 0·1691 thermal units. With these data the ratio of the specific heats, K_p/K_v or γ, is therefore 1·404. Determinations of γ in air by other methods have generally given a slightly higher value, such as 1·408. For most calculations 1·4 is near enough.

To find the constants for other gases, take values of c inversely proportional to the density as given in Regnault's Tables[1]. These tables also give values of K_p. K_v is then found by subtraction of c from K_p.

37. Work done by an expanding fluid. We now return to the consideration of imaginary indicator diagrams, which exhibit the relation of the pressure to the volume of a fluid working substance during its expansion or during its compression, in order to study the form which the expansion or compression curve assumes in certain particular cases.

In most of the instances which present themselves in the theory of heat-engines such curves may be exactly or approximately represented by an equation of the form

$$PV^n = \text{constant},$$

where the index n has various numerical values but is a constant for any one curve. We proceed to find the values which n takes in two very important modes of expansion. Let AB, figure 10, be a curve of expansion, for any fluid, to which the general formula $PV^n=$ constant is applicable. The fluid is supposed to expand from A, where the pressure is P_1 and the volume V_1, to B, where the pressure is P_2 and the volume is V_2.

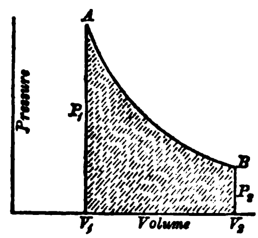


FIG. 10.

During this expansion it does an amount of work which is measured by the shaded area under the curve. That is to say, if W denote the work done during expansion,

$$W = \int_{V_1}^{V_2} PdV \text{..........................} (4),$$

[1] See Everett's *Units and Physical Constants.*

the integral being taken between the limits V_2 and V_1. To integrate we have to remember that the pressure and volume at any point are such that

$$PV^n = P_1V_1^n = P_2V_2^n.$$

Hence substituting $\frac{P_1V_1^n}{V^n}$ for P in (4), this expression for the work done becomes

$$W = P_1V_1^n \int_{V_1}^{V_2} \frac{dV}{V^n},$$

which gives on integration

$$W = \frac{P_1V_1^n(V_2^{1-n} - V_1^{1-n})}{1-n} \quad \text{.................(5).}$$

This may also be written

$$W = \frac{P_1V_1(1 - r^{1-n})}{n-1}, \quad \text{..........................(6)}$$

where r is the *ratio of expansion*, that is to say, the ratio of the final volume V_2 to the initial volume V_1.

Since $P_1V_1^n = P_2V_2^n$, still another form in which the above result may be expressed is readily derived from equation (5), namely,

$$W = \frac{P_1V_1 - P_2V_2}{n-1} \quad \text{..........................(7).}$$

If instead of expanding from A to B the fluid were compressed from B to A the expression given above for W will measure the work spent upon the fluid instead of work done by it.

Further, if the working substance be a gas, in which (by the laws of Boyle and Charles) $PV = c\tau$, equation (7) may be written

$$W = \frac{c(\tau_1 - \tau_2)}{n-1} \quad \text{............................(8),}$$

since $P_1V_1 = c\tau_1$ and $P_2V_2 = c\tau_2$.

38. Adiabatic Expansion. We have next to consider particular modes in which any working substance may be expanded or compressed. One very important case is that which occurs when the fluid neither receives nor rejects heat as it expands, or as it is compressed. This mode of expansion or compression is called *adiabatic*, and a curve which exhibits the relation of P to V in such a process is called an adiabatic

line. In any adiabatic process the substance is neither gaining nor losing heat by conduction or radiation or internal chemical action. Hence the work which a substance does when it is expanding adiabatically is all done at the expense of its stock of internal energy, and the work which is spent upon a substance when it is being compressed adiabatically all goes to increase its stock of internal energy. Adiabatic action would be realized if we had a substance expanding, or being compressed, without chemical change, in a cylinder which (along with the piston) was a perfect non-conductor of heat, and was opaque to heat-rays.

In actual heat-engines the action is never strictly adiabatic on account of the fact that more or less heat passes by conduction between the working fluid and the inner surface of the cylinder. The more quickly the process of expansion or compression is performed the more nearly adiabatic it becomes, for there is then less time for this transfer of heat to take place.

Coming now to the particular case in which the working substance is a gas, since in adiabatic expansion or compression the work done is equal to the change of internal energy we may determine the law of adiabatic action in a gas as follows. Taking expression (8) for the work done, namely,

$$W = \frac{c(\tau_1 - \tau_2)}{n-1}$$

we have to find what value of n in the general formula $PV^n =$ constant will make the process adiabatic. We have seen (§ 34) that in any change of temperature from τ_1 to τ_2 a gas loses internal energy to the amount

$$K_v(\tau_1 - \tau_2),$$

which may be written

$$\frac{c(\tau_1 - \tau_2)}{\gamma - 1},$$

γ being (§ 35) the ratio of the two specific heats.

Hence, equating the work done with the loss of internal energy, the condition of adiabatic expansion is secured when

$$\frac{c(\tau_1 - \tau_2)}{n-1} = \frac{c(\tau_1 - \tau_2)}{\gamma - 1} \quad \text{.....................(9)},$$

from which $$n = \gamma.$$

Expansion or compression will therefore be adiabatic when

$$PV^\gamma = \text{constant}, \qquad\qquad (10),$$

or in other words this is the equation of an adiabatic line for a gas.

39. Change of temperature in the adiabatic expansion of a gas. When a gas is expanding adiabatically its stock of internal energy is being reduced, and hence its temperature (to which the internal energy is proportional, by § 34) falls. Conversely, in adiabatic compression the temperature rises. The amount by which the temperature changes is found by combining the equations

$$P_1V_1^\gamma = P_2V_2^\gamma \text{ and } \frac{P_2V_2}{P_1V_1} = \frac{\tau_2}{\tau_1}.$$

Multiplying them together we have

$$\frac{\tau_2}{\tau_1} = \frac{P_2V_2P_1V_1^\gamma}{P_1V_1P_2V_2^\gamma}$$

whence

$$\left.\begin{aligned} \frac{\tau_2}{\tau_1} &= \left(\frac{V_1}{V_2}\right)^{\gamma-1} \\ \text{or} \quad \frac{\tau_2}{\tau_1} &= \left(\frac{1}{r}\right)^{\gamma-1} \end{aligned}\right\} \qquad\qquad (11),$$

where r is as before the ratio of expansion. This result of course applies to compression as well as to expansion along an adiabatic line.

As to expansions which are not adiabatic, it follows, from the expressions given above for the external work done by an expanding gas and for the change of internal energy, that if n is less than γ the work done is greater than the loss of internal energy—that is to say, the gas is then taking in heat while it expands. On the other hand, if n is greater than γ the work done is less than the loss of internal energy; in other words, the gas is then rejecting heat by conduction to the walls of the containing vessel or in some other way.

By way of exemplifying an adiabatic process suppose a quantity of dry air to be contained in a cylinder at a temperature of 60° Fah. ($\tau = 521$) and to be suddenly compressed to half its original volume, the process being so rapid that no appreciable part of the heat developed by compression has time to pass from the air to the cylinder walls. Here $r = \frac{1}{2}$, and taking γ for air to

be 1·404 the temperature immediately after compression, before the gas has time to cool, is

$$\tau_2 = \tau_1 \left(\frac{1}{r}\right)^{\gamma-1} = 521 \times 2^{0.404} = 689$$

or 228° Fah. The work spent in compressing the air, namely,

$$\frac{c(\tau_2 - \tau_1)}{\gamma - 1} \text{ is } \frac{53{\cdot}18 \times 168}{0{\cdot}404} = 22110 \text{ foot-pounds},$$

for each lb. of air in the cylinder. The internal energy of the gas becomes increased by this amount; but if the cylinder be a conductor of heat the whole of this will in time become dissipated by conduction to surrounding bodies and the internal energy will gradually return to its original value, as the temperature of the gas sinks to 60° Fah.

During compression the pressure rises (following the law PV^γ = constant), and just at the end its value is greater than at the beginning in the ratio r^γ to 1, that is $2^{1.404}$ or 2·65 to 1. If as before we suppose the temperature to sink slowly by conduction to 60° Fah. while the volume does not change, the pressure will fall with the temperature until it reaches a value only twice that which it had before the air was compressed.

40. Isothermal Expansion. Another very important mode of expansion or compression is that called *isothermal*, in which the temperature of the working substance is kept constant during the process.

In the case of a gas the curve of isothermal expansion is a rectangular hyperbola, having the equation

$$PV = \text{constant} = c\tau \ldots\ldots\ldots\ldots\ldots\ldots (12).$$

This is a particular case of the general formula PV^n = constant. But equation (6) or (7) above will not serve to find the work done, for when $n = 1$ both the numerator and the denominator in these expressions vanish. To find the work done in the isothermal expansion of a gas we have

$$W = \int_{V_1}^{V} PdV$$

and $$P = \frac{P_1 V_1}{V}$$

from which $$W = P_1 V_1 \int_{V_1}^{V_2} \frac{dV}{V}.$$

Integrating, $$W = P_1V_1 (\log_e V_2 - \log_e V_1)$$

or $$W = P_1V_1 \log_e \frac{V_2}{V_1} = P_1V_1 \log_e r \text{ ......} \quad (13).$$

Instead of P_1V_1 we may write PV, since the product of P and V is constant through the process, and again, since $PV = c\tau$,

$$W = c\tau \log_e r \text{.......................} \quad (14).$$

There is no need here to use a suffix with τ since the temperature does not change. These expressions give either the work done by a gas during isothermal expansion or the work spent upon it during isothermal compression[1].

During isothermal expansion or compression a gas suffers no change of internal energy (by § 34, since τ is constant). Hence during isothermal expansion the gas must take in an amount of heat just equal to the work it does, and during isothermal compression it must reject an amount of heat just equal to the work spent upon it. The expression $c\tau \log_e r$ consequently measures, not only the work done by or upon the gas, but also the heat taken in during isothermal expansion or given out during isothermal compression. In the diagram, fig. 11, the line AB is an example of a curve of isothermal expansion for a perfect gas, called for brevity an isothermal line, while AC is an adiabatic line starting from the same point A.

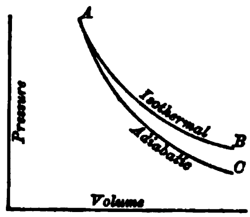


FIG. 11.

The compression of air or any other gas in a real cylinder is approximately adiabatic when the process is very quickly performed, but approximately isothermal when it is performed so slowly that the heat has time to be dissipated by conduction while the process goes on.

41. Carnot's Cycle of operations. We shall now consider the action of an ideal engine in which the working substance is a perfect gas, that is, caused to pass through a cycle of changes each

[1] In calculations where this expression is involved it is convenient to remember that $\log_e$, the 'hyperbolic,' or 'natural,' or 'Napierian' logarithm, of any number is 2·3026 times the common logarithm of the number.

of which is either isothermal or adiabatic. The cycle to be described was first examined by Carnot, and is spoken of as

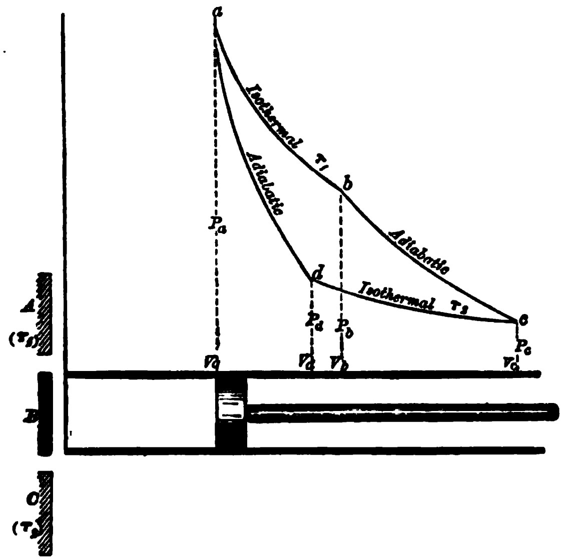


FIG. 12. Carnot's Cycle, with a gas for working substance.

Carnot's cycle of operations. Imagine a cylinder and piston composed of a perfectly non-conducting material, except as regards the bottom of the cylinder, which is a conductor. Imagine also a hot body or indefinitely capacious source of heat *A*, kept always at a temperature τ_1, also a perfectly non-conducting cover *B*, and a cold body or indefinitely capacious receiver of heat *C*, kept always at some temperature τ_2 which is lower than τ_1. It is supposed that *A*, *B*, or *C* can be applied at will to the bottom of the cylinder. Let the cylinder contain 1 lb. of a perfect gas, at temperature τ_1, volume V_a, and pressure P_a to begin with. The suffixes refer to the points on the indicator diagram, fig. 12.

(1) Apply *A*, and allow the piston to rise slowly through any convenient distance. The gas expands isothermally at τ_1, taking in heat from the hot source *A* and doing work. The pressure changes to P_b and the volume to V_b.

(2) Remove *A* and apply *B*. Allow the piston to go on rising. The gas expands adiabatically, doing work at the expense of its internal energy, and the temperature falls. Let this go on

until the temperature is τ_2. The pressure is then P_c, and the volume V_c.

(3) Remove B and apply C. Force the piston down slowly. The gas is compressed isothermally at τ_2, since the smallest increase of temperature above τ_2 causes heat to pass into C. Work is spent upon the gas, and heat is rejected to the cold receiver C. Let this be continued until a certain point d (fig. 12) is reached, such that the fourth operation will complete the cycle.

(4) Remove C and apply B. Continue the compression, which is now adiabatic. The pressure and temperature rise, and if the point d has been properly chosen, when the pressure is restored to its original value P_a, the temperature will also have risen to its original value τ_1. [In other words, the third operation must be stopped when a point d is reached such that an adiabatic line drawn through d will pass through a.] This completes the cycle.

To find the proper place at which to stop the third operation, we have by equation (11), for the cooling during the adiabatic expansion of stage (2),

$$\frac{\tau_1}{\tau_2} = \left(\frac{V_c}{V_b}\right)^{\gamma-1}$$

and also, for the heating during the adiabatic compression of stage (4),

$$\frac{\tau_1}{\tau_2} = \left(\frac{V_d}{V_a}\right)^{\gamma-1}.$$

Hence $$\frac{V_c}{V_b} = \frac{V_d}{V_a},$$

and therefore also $$\frac{V_c}{V_d} = \frac{V_b}{V_a}.$$

That is to say, the ratio of isothermal compression in the third stage of the cycle is to be made equal to the ratio of isothermal expansion in the first stage, in order that an adiabatic line through d shall complete the cycle. For brevity we shall denote either of these last ratios (of isothermal expansion and compression) by r.

The following are the transfers of heat to and from tho working gas, in the four successive stages of the cycle:—

(1) Heat taken in from $A = c\tau_1 \log_e r$ (by § 40).

(2) No heat taken in or rejected.

(3) Heat rejected to $C = c\tau_2 \log_e r$ (by § 40).

(4) No heat taken in or rejected.

Hence, the net amount of external work done by the gas, being the excess of the heat taken in above the heat rejected in a complete cycle, is

$$c(\tau_1 - \tau_2)\log_e r;$$

this is the area enclosed by the four curves in Fig. 12.

42. Efficiency in Carnot's Cycle. The *efficiency* of the process, namely, the fraction

$$\frac{\text{Heat converted into work}}{\text{Heat taken in}}$$

is

$$\frac{c(\tau_1 - \tau_2)\log_e r}{c\tau_1 \log_e r} = \frac{\tau_1 - \tau_2}{\tau_1} \quad \text{.................. (15).}$$

This is the fraction of the whole heat given to it which an engine following Carnot's cycle converts into work. The engine takes in an amount of heat, at the temperature of the source, proportional to τ_1; it rejects an amount of heat, at the temperature of the receiver, proportional to τ_2. It works within a range of temperature extending from τ_1 to τ_2, by letting down heat from τ_1 to τ_2 (§ 25), and in the process it converts into work a fraction of that heat, which fraction will be greater the lower the temperature τ_2 at which heat is rejected is below the temperature τ_1 at which heat is received.

43. Carnot's Cycle reversed. Next consider what will happen if we reverse Carnot's cycle, that is to say, if we force this imaginary engine to act so that the same indicator diagram as before is traced out, but in the direction opposite to that followed in § 41. Starting as before from the point a (fig. 12) and with the gas at τ_1, we shall require the following four operations:—

(1) Apply B and allow the piston to rise. The gas expands adiabatically, the curve traced is ad, and when d is reached the temperature has fallen to τ_2.

(2) Remove B and apply C. Allow the piston to go on rising. The gas expands isothermally at τ_2, taking heat from C and the curve dc is traced.

(3) Remove C and apply B. Compress the gas. The process is adiabatic. The curve traced is cb, and when b is reached the temperature has risen to τ_1.

(4) Remove B and apply A. Continue the compression, which is now isothermal, at τ_1. Heat is now rejected to A, and the cycle is completed by the curve ba.

In this process the engine is not doing work; on the contrary, a quantity of work is spent upon it equal to the area of the diagram, or $c(\tau_1-\tau_2)\log_e r$, and this work is converted into heat. Heat is taken in from C in the first operation, to the amount $c\tau_2\log_e r$. Heat is rejected to A in the fourth operation, to the amount $c\tau_1\log_e r$. In the first and third operations there is no transfer of heat.

The action is now in every respect the reverse of what it was before. The same work is now spent upon the engine as was formerly done by it. The same amount of heat is now given to the hot body A as was formerly taken from it. The same amount of heat is now taken from the cold body C as was formerly given to it, as will be seen by the following scheme:—

Carnot's Cycle, Direct.

Work done by the gas $= c(\tau_1-\tau_2)\log_e r$;
Heat taken from $A = c\tau_1\log_e r$;
Heat rejected to $C = c\tau_2\log_e r$.

Carnot's Cycle, Reversed.

Work spent upon the gas $= c(\tau_1-\tau_2)\log_e r$;
Heat rejected to $A = c\tau_1\log_e r$;
Heat taken from $C = c\tau_2\log_e r$.

The reversal of the work has been accompanied by an exact reversal of each of the transfers of heat.

44. Reversible engine. An engine in which this is possible is called, from the thermodynamic point of view, a *reversible* engine. In other words, a reversible heat-engine is one which, if forced to trace out its indicator diagram reversed in direction, so that the work which would be done by the engine, when running direct, is actually spent upon it, will reject to the source of heat the same quantity of heat as, when running direct, it would take from the source, and will take from the receiver of heat the same quantity as, when running direct, it would reject to the receiver. By "the source of heat" is meant the hot body which acts as

source when the engine is running direct, and by "the receiver" is meant the cold body which then acts as receiver. An engine performing Carnot's cycle of operations is one example of a reversible engine. The idea of thermodynamic reversibility in the sense here defined is of the greatest interest, for the reason that no heat-engine can be more efficient than a reversible engine when both work between the same limits of temperature; that is to say, when both engines take in heat at the same temperature and also reject heat at the same temperature. This theorem, due to Carnot, is of fundamental importance in the theory of heat-engines. It is deduced as follows from the laws of thermodynamics.

45. Carnot's Principle. To prove that no other heat-engine can be more efficient than a reversible engine when both work between the same limits of temperature, imagine two engines R and S of which R is reversible, and let them work by taking in heat from a hot body A and by rejecting heat to a cold body C. Let Q_A be the quantity of heat which the reversible engine R takes in from A for each unit of work which it does and let Q_C be the quantity which it rejects to C.

Now consider what consequences would follow if it were possible for S to be more efficient than R. It would take in less heat from A and reject correspondingly less heat to C, in doing each unit of work. Denote the heat which it would take in from A by $Q_A - q$ and the heat which it would reject to C by $Q_C - q$.

Suppose that S working direct (that is to say, converting heat into work) be set to drive R as a reversed engine, so that R converts work into heat. For every unit of work done by the engine S on the reversible engine R the quantity $Q_A - q$ would be taken from A by the engine S, and the quantity Q_A would be restored to A by the reversed action of the engine R. This is because R being reversible restores to A when working reversed the same amount of heat as it would take from A when working direct. Hence the hot body would on the whole gain heat, by the amount q for every unit of work done by the one engine on the other. Again, S gives to C a quantity $Q_C - q$ while R takes from C a quantity Q_C and hence the cold body C would lose an amount of heat equal to q for every unit of work done by the one engine or the other. Thus the combined action of the two engines—one

working direct, as a true heat-engine, and the other reversed, as what we might call a heat-pump—would result in a transfer of heat from the cold body C to the hot body A, and this process might evidently go on without limit. Moreover the two engines taken together form a purely self-acting system, for the whole power generated in one is spent on the other and is sufficient to drive the other; if we assume that there is no mechanical friction the double machine requires no help from without. Hence the supposition that the engine S could be more efficient than the reversible engine R has led to a result inconsistent with the second law of thermodynamics for it has led us to construct, in imagination, a self-acting machine capable of transferring heat, in any quantity, from a cold body to a hot body. The second law asserts that this is contrary to all experience, and we are therefore forced to the conclusion that no other engine S can be more efficient than a reversible engine R when both work between the same limits of temperature. In other words, when the source and receiver of heat are given a reversible heat-engine is as efficient as any engine working between them can be.

Further, let both engines be reversible. Then the same argument shows that neither can be more efficient than the other. Hence all reversible heat-engines taking in and rejecting heat at the same two temperatures are equally efficient.

46. Reversibility the criterion of perfection in a heat-engine. These results imply that reversibility, in the thermodynamic sense, is the criterion of what may be called perfection in a heat-engine. A reversible engine is perfect in the sense that it cannot be improved on as regards efficiency: no other engine, taking in and rejecting heat at the same temperatures, will convert into work a greater fraction of the heat which it takes in. Moreover, if this criterion be satisfied, it is as regards efficiency a matter of complete indifference what is the nature of the working substance, or what, in other respects, is the mode of the engine's action.

47. Efficiency of a perfect heat-engine. Further, since all engines that are reversible are equally efficient, provided they work between the same temperatures, an expression for the efficiency of one will apply equally to all. Now, the engine whose

efficiency was found in § 42, namely, an engine having a gas for working substance and performing Carnot's Cycle of operations, is one example of a reversible engine. Hence the expression which was obtained for its efficiency, namely,

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

is the efficiency of any reversible heat-engine whatsoever taking in heat at τ_1 and rejecting heat at τ_2. And, as no engine can be more efficient than one that is reversible, this expression is the measure of *perfect efficiency*. We have thus arrived at the immensely important conclusion that no heat-engine can convert into work a greater fraction of the heat which it receives than is expressed by the excess of the temperature of reception above that of rejection divided by the absolute temperature of reception.

48. Summary of the argument. Briefly recapitulated, the steps of the argument by which this result has been reached are as follows. After stating the experimental laws to which gases conform and finding that they afforded a provisional means of defining temperature upon an absolute scale we examined the action of a heat-engine in which the working substance took in heat when at the temperature of the source and rejected heat when at the temperature of the receiver, the change of temperature from one to the other of these limits being accomplished by adiabatic expansion and adiabatic compression. Taking a special case in which the engine had for its working substance a perfect gas, we found that its efficiency was $(\tau_1 - \tau_2)/\tau_1$ (§ 42). We also observed that it was, in the thermodynamic sense, a reversible engine (§ 44). Then we found, by an application of the second law of thermodynamics, that no heat-engine can have a higher efficiency than a reversible engine, when taking in and giving out heat at the same two temperatures τ_1 and τ_2; this was shown by the fact that a contrary assumption would lead to a violation of the second law (§ 45). Hence, we concluded that all reversible heat-engines receiving and rejecting heat at the same temperatures, τ_1 and τ_2 respectively, are equally efficient, and hence that the efficiency

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

already determined for one particular reversible engine, is the

efficiency of any reversible engine, and is a limit of efficiency which no engine whatever can exceed.

Another way of stating the performance of a perfect engine evidently is to say that the heat taken in Q_1 is to the heat rejected Q_2 as τ_1 is to τ_2, or

$$\frac{Q_1}{\tau_1} = \frac{Q_2}{\tau_2} \quad \text{.......................... (16).}$$

49. Conditions of maximum efficiency. The availability of heat for transformation into work depends essentially on the range of temperature through which the heat is let down from that of the hot source to that of the cold body into which heat is rejected; it is only in virtue of a difference of temperature between bodies that conversion of any part of their heat into work becomes possible. No mechanical effect could be produced from heat, however great the amount of heat obtainable, if all bodies were at a dead level of temperature. Again, it is impossible to convert the whole of any supply of heat into work because it is impossible to reach the absolute zero of temperature at the lower end of the temperature range.

If τ_1 and τ_2 are given as the highest and lowest temperatures of the range through which a heat-engine is to work, it is clear that the maximum of efficiency can be reached only when the engine takes in all its heat at τ_1 and rejects at τ_2 all that is rejected. With respect to every portion of heat taken in and rejected the greatest ideal efficiency is

$$\frac{\text{Temperature of reception} - \text{temperature of rejection}}{\text{Temperature of reception}}.$$

Any heat taken in at a temperature below τ_1 or rejected at a temperature above τ_2 will have less availability for conversion into work than if it had been taken in at τ_1 and rejected at τ_2, and hence, with a given pair of limiting temperatures, it is essential to maximum efficiency that no heat be taken in by the engine except at the top of the range, and no heat rejected except at the bottom of the range. Further, as we have seen in § 45, when the temperatures at which heat is received and rejected are assigned, an engine attains the maximum of efficiency if it be reversible.

50. Conditions of reversibility. It is therefore important to inquire more particularly what kinds of action are reversible in

the thermodynamic sense. A little consideration will show that a transfer of heat from the source to the working substance, or from the working substance to the receiver, is reversible only when the working substance is at sensibly the same temperature as the source or the receiver, as the case may be, and an expansion is reversible only when it occurs by the gradual displacement of some part of the containing envelope in such a manner that the expanding fluid does external work on the envelope, and does not waste energy to any sensible extent in setting itself in motion. This excludes what may be termed free expansion, such as that of the gas in Joule's experiment, § 34, and it excludes also what may be called imperfectly-resisted expansion, such as would occur if the fluid were allowed to expand into a chamber in which the pressure was less than that of the fluid, or if the fluid were expanding in a cylinder under a piston which rose so fast as to cause, through the inertia of the expanding fluid, local variations of pressure throughout the cylinder. A similar condition of course applies in regard to the compression of the working fluid: neither expansion nor compression must take place in such a manner as to set up eddies within the fluid.

To make a heat-engine, working within given limits of temperature, as efficient as possible the conditions to aim at therefore are—(1) to take in no heat except at the highest temperature, and to reject no heat except at the lowest temperature, (2) to secure that the working substance shall, when receiving heat, be at the temperature of the body from which the heat comes, and that it shall, when giving up heat, be at the temperature of the body to which heat is given up; (3) to avoid free or imperfectly-resisted expansion. If these conditions are fulfilled the engine is a reversible heat-engine and the most efficient possible within the given range of temperatures.

The first and second of these conditions are satisfied if in the action of the engine the working substance changes its temperature from τ_1 to τ_2 by adiabatic expansion, and from τ_2 to τ_1 by adiabatic compression, thereby being enabled to take in and reject heat at the ends of the range without taking in or rejecting any by the way. This is the action in Carnot's ideal engine (§ 41).

51. Perfect Engine using Regenerator. But there is another way in which the action of a heat-engine may be made

reversible. Suppose that the working substance can be caused to deposit heat in some body within the engine while passing from τ_1 to τ_2, in such a manner that the transfer of heat from the substance to this body is reversible (satisfying the second condition above), then when we wish the working substance to pass from τ_2 to τ_1 we may reverse this transfer and so recover the heat that was deposited in this body. This alternate storing and restoring of heat would serve, instead of adiabatic expansion and compression, to make the temperature of the working substance pass from τ_1 to τ_2 and from τ_2 to τ_1 respectively. The alternate storing and restoring is an action occurring wholly within the engine, and is therefore distinct from the taking in and rejecting of heat by the engine.

In 1827 Robert Stirling designed an apparatus, called a *regenerator*, by which this process of alternate storing and restoring of heat could be actually performed. For the present purpose it will suffice to describe the regenerator as a passage through which the working fluid can travel in either direction, whose walls have a very large capacity for heat, so that the amount alternately given to or taken from them by the working fluid causes no more than an insensible rise or fall in their temperature. The temperature of the walls at one end of the passage is τ_1, and this falls continuously down to τ_2 at the other end. When the working fluid at temperature τ_1 enters the hot end and passes through, it comes out at the cold end at temperature τ_2, having stored in the walls of the regenerator a quantity of heat which it will pick up again when passing through in the opposite direction. During the return journey of the working fluid through the regenerator from the cold to the hot end its temperature rises from τ_2 to τ_1 by picking up the heat which was deposited when the working fluid passed through from the hot end to the cold. The process is strictly reversible, or rather would be so if the regenerator had an unlimited capacity for heat, if no conduction of heat took place along its walls from the hot to the cold end, and if no loss took place by conduction or radiation from its external surface. Such a regenerator is of course an ideal impossible to realise in practice.

52. Stirling's Regenerative Air-Engine. Using air as the working substance, and employing his regenerator, Stirling made an engine (to be described later) which, allowing for

practical imperfections, is the earliest example of a reversible engine. The cycle of operations in Stirling's engine was substantially this:

(1) Air (which had been heated to τ_1 by passing through the regenerator) was allowed to expand isothermally through a ratio r, taking in heat from a furnace and raising a piston. Heat taken in (per lb. of air) $= c\tau_1 \log_e r$.

(2) The air was caused to pass through the regenerator from the hot to the cold end, depositing heat and having its temperature lowered to τ_2, without change of volume. Heat stored in regenerator $= K_v(\tau_1 - \tau_2)$. The pressure of course fell in proportion to the fall in temperature.

(3) The air was then compressed isothermally to its original volume at τ_2 in contact with a refrigerator (or receiver of heat). Heat rejected $= c\tau_2 \log_e r$.

(4) The air was again passed through the regenerator from the cold to the hot end, taking up heat and having its temperature raised to τ_1. Heat restored by the regenerator $= K_v(\tau_1 - \tau_2)$. This completed the cycle.

The efficiency is

$$\frac{c\tau_1 \log_e r - c\tau_2 \log_e r}{c\tau_1 \log_e r} = \frac{\tau_1 - \tau_2}{\tau_1}.$$

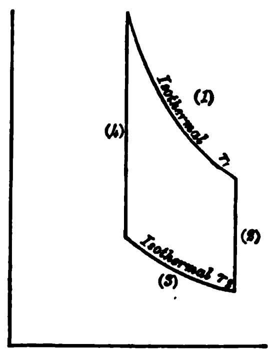


FIG. 13. Ideal Indicator diagram of Air-Engine with Regenerator (Stirling).

The indicator diagram of this action is shown in fig. 13. Stirling's engine is important, not as a present-day heat-engine (though it has recently been revived in small forms after a long interval of disuse), but because it is typical of the only mode, other than Carnot's plan of adiabatic expansion and compression, by which the action of a heat-engine can be made reversible.

The regenerative principle has been largely used in metallurgy and other industrial processes: the Siemens steel-furnace is an example of its application on a large scale. Notwithstanding the immensely valuable services which the regenerator has rendered in such processes, its application to heat-engines has hitherto been

very limited. Another way of using it in air-engines was tried by Ericsson, who kept the pressure instead of the volume constant while the working air was passed through the regenerator, thus getting an indicator diagram consisting of two isothermal lines and two lines of equal pressure. Attempts have also been made by Siemens and by Fleeming Jenkin to apply it to steam-engines and to gas-engines. It has also found application in some refrigerating machines, or reversed heat-engines, to which reference will be made in a later chapter. But almost all actual heat-engines, in so far as they can be said to approach the condition of reversibility, do so, not by the use of the regenerative principle, but by more or less nearly adiabatic expansion and compression after the manner of Carnot's ideal engine.

CHAPTER III.

PROPERTIES OF STEAM AND ELEMENTARY THEORY OF THE STEAM-ENGINE.

53. Formation of steam under constant pressure. We have now to consider the action of heat-engines in which the working substance is water and water-vapour or steam, and as a preliminary to this it is necessary to give some account of the physical properties of steam as determined by experiment. The properties of steam are most conveniently stated by referring in the first instance to what happens when steam is formed *under constant pressure.* This is substantially the process which occurs in the boiler of a steam-engine when the engine is at work. To fix the ideas we may suppose that the vessel in which steam is to be formed is a long upright cylinder fitted with a frictionless piston which may be loaded so that it exerts a constant pressure on the fluid below. Let there be, to begin with, at the foot of the cylinder a quantity of water (which for convenience of numerical statement we shall take as 1 lb.), at any temperature t_0; and let the piston press on the surface of the water with a force of P lbs. per square foot. Let heat now be applied to the bottom of the cylinder. As it enters the water it will produce the following effects in three stages:—

(1) The temperature of the water rises until a certain temperature t is reached, at which steam begins to be formed. The value of t depends on the particular pressure P which the piston exerts. Until the temperature t is reached there is nothing but water below the piston.

(2) Steam is formed, more heat being taken in. The piston (which is supposed to exert a constant pressure) rises. No further increase of temperature occurs during this stage, which continues until all the water is converted into steam. During this stage the steam which is formed is said to be *saturated*. The volume which the piston encloses at the end of this stage,—the volume, namely, of 1 lb. of saturated steam at pressure P (and temperature t),—will be denoted by V in cubic feet.

(3) If after all the water is converted into steam more heat be allowed to enter, the volume will increase and the temperature will rise. The steam is then said to be *superheated*.

54. Saturated and superheated steam. The difference between saturated and superheated steam may be expressed by saying that if water (at the temperature of the steam) be mixed with steam some of the water will be evaporated if the steam is superheated, but none of the steam is saturated. Any vapour in contact with its liquid and in thermal equilibrium is necessarily saturated. When saturated its properties differ considerably, as a rule, from those of a perfect gas, but when superheated they approach those of a perfect gas more and more closely the farther the process of superheating is carried, that is to say, the more the temperature is raised above t, the temperature of saturation corresponding to the given pressure P. Saturated steam at a given pressure can have but one temperature; superheated steam at the same pressure can have any temperature higher than that.

55. Relation of pressure and temperature in saturated steam. The temperature t at which steam is formed depends on the value of P. The relation of pressure to temperature was determined with great care by Regnault, in a series of classical experiments on which our knowledge of the properties of steam chiefly depends[1]. The pressure of saturated steam rises with the temperature at a rate which increases rapidly in the upper regions of the scale. This will be apparent from the first and second columns of Table I, given on p. 67, which is compiled from Rankine's reduction of Regnault's results. The first column gives

[1] *Mem. Inst. France*, 1847, vol. xxi. An account of Regnault's methods of experiment and a statement of his results expressed in British measures will be found in Dixon's *Treatise on Heat* (Dublin, 1849).

the temperature on the Fahr. scale; the second gives the corresponding pressure in pounds per square inch. Rankine has also expressed the relation of temperature and pressure in saturated steam by the following formula (which is applicable with other constants to other vapours[1]):—

$$\log p = 6{\cdot}1007 - \frac{2732}{\tau} - \frac{396945}{\tau^2} \ldots\ldots\ldots\ldots(1),$$

where p is the pressure in pounds per square inch, and τ is the absolute temperature in Fahr. degrees. For most purposes, however, it is more convenient to find the pressure corresponding to a given temperature, or the temperature corresponding to a given pressure, from the table, either by interpolation or by drawing a portion of the curve connecting P with t. A more extended table will be found in the Appendix, where the temperature of saturated steam, to the nearest half degree, is given for various pressures.

56. Relation of pressure and volume in saturated steam. Table I. also shows the volume V, in cubic feet, occupied by 1 lb. of saturated steam at each pressure, and a more extensive series of values are given in the table in the Appendix. The volume of a pound of saturated steam at any assigned pressure is a quantity difficult to measure by direct experiment. It may, however, be calculated, from a knowledge of other properties of steam, by a process which will be described in the next chapter (§ 74). The values of V given in the table were determined by means of this process; they agree fairly well with such direct observations of the density of steam as have hitherto been made[2].

The relation of P to V may be approximately expressed by the formula[3]

$$PV^{\frac{17}{16}} = \text{constant} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2),$$

[1] *Phil. Mag.* Dec. 1854, or *Manual of the Steam-Engine*, p. 237.

[2] The values of V given in the table are found from those given by Rankine in his treatise on the Steam-Engine. He employed the method of calculating V alluded to in the text, but the numbers which he gave require alteration in consequence of the fact that J, the mechanical equivalent of heat, enters into the formula (see § 75); the calculated values of V are in fact nearly proportional to J. In Rankine's calculation J was taken as 772: since the number 778 has been adopted here the values of V given in the table are increased nearly in the ratio of 778 to 772.

[3] This is Rankine's formula where the index is expressed in a form convenient for logarithmic calculation. Zeuner considers that the curve of P and V for saturated steam is better expressed by using a slightly different index, and gives the equation $PV^{1{\cdot}0646} = \text{constant}$. (*Technische Thermodynamik*, Vol. II. p. 36.)

Temperature on Fahrenheit scale.	Pressure.	Volume of 1 lb.	Heat of Formation.	
			H.	*h.*
Degrees.	Lb. per sq. in.	Cub. Ft.	Thermal Units.	Thermal Units.
32	0·085	3416	1091·8	0
41	0·122	2425	1094·5	9·0
50	0·173	1745	1097·3	18·0
59	0·241	1274	1100·0	27·0
68	0·333	942	1102·8	36·0
77	0·452	705	1105·5	45·0
86	0·607	533	1108·2	54·0
95	0·806	408	1111·0	63·0
104	1·06	315	1113·7	72·0
113	1·38	245	1116·5	81·0
122	1·78	193·5	1119·2	90·1
131	2·27	153·6	1121·9	99·1
140	2·88	123·0	1124·7	108·1
149	3·62	99·2	1127·4	117·1
158	4·51	80·6	1130·2	126·2
167	5·58	66·1	1132·9	135·2
176	6·86	54·34	1135·6	144·3
185	8·38	45·05	1138·4	153·3
194	10·16	37·55	1141·1	162·4
203	12·26	31·50	1143·9	171·4
212	14·70	26·56	1146·6	180·5
221	17·53	22·51	1149·3	189·6
230	20·80	19·18	1152·1	198·7
239	24·54	16·40	1154·8	207·8
248	28·83	14·11	1157·6	216·9
257	33·71	12·18	1160·3	226·0
266	39·25	10·56	1163·1	235·2
275	45·49	9·195	1165·8	244·3
284	52·52	8·035	1168·6	253·5
293	60·40	7·046	1171·3	262·7
302	69·21	6·201	1174·1	271·9
311	79·03	5·475	1176·8	281·1
320	89·86	4·853	1179·5	290·3
329	101·9	4·317	1182·2	299·5
338	115·1	3·843	1185·0	308·7
347	129·8	3·437	1187·7	318·0
356	145·8	3·081	1190·4	327·3
365	163·3	2·770	1193·2	336·6
374	182·4	2·495	1195·9	345·9
383	203·3	2·253	1198·6	352·2
392	225·9	2·041	1201·4	364·5
401	250·3	1·852	1204·1	373·9
410	276·9	1·685	1206·9	383·2
419	305·5	1·537	1209·6	392·6
428	336·3	1·404	1212·4	402·0

the value of the constant (if J be taken as 778) being about 69000 when P is stated in lbs. per sq. foot and V in cub. feet per lb.

The student will find it useful to draw curves, with the data of the table, showing the relation between the pressure and the temperature of saturated steam and also the relation of pressure to volume $\left(\text{or to the density, which is } \frac{1}{V}\right)$, especially within the range usual in steam-engine practice. He will observe that $\frac{dP}{dt}$, the rate of change of pressure with respect to change of temperature increases rapidly as the temperature rises, and hence that in the upper part of the range a very small elevation of temperature in a boiler is necessarily associated with a large increment of pressure. The familiar case of water boiling in a kettle or other open vessel is only a special case of the formation of steam under constant pressure. There the constant pressure is that of the atmosphere, which is 14·7 lbs. per square inch or thereabouts (as indicated by the barometer) and consequently the temperature at which the water boils is about 212° F.

57. Supply of heat in the formation of steam under constant pressure. We have next to consider the supply of heat in the imaginary experiment of § 53 in which 1 lb. of water initially at some temperature t_0 is first heated to the boiling point and then converted into steam, under a constant pressure P, this constant pressure determining what the temperature of the boiling point shall be. During the first stage, while the temperature is rising from its initial value t_0 to t, no steam is formed, and heat is required only to warm the water. Since the specific heat of water is nearly constant, the amount of heat taken in during the first stage is approximately $t - t_0$ thermal units or $J(t - t_0)$ foot-pounds, and this expression will generally serve with sufficient accuracy in practical calculations. More exactly, however, the heat taken in is in general somewhat greater than this, for Regnault's experiments show that the specific heat of water increases slightly at high temperatures. In stating the amount of heat required for this first stage, t_0 must be taken as a known temperature; for convenience in numerical statement the temperature 32° F. is usually chosen as an arbitrary starting-point from which the reception of heat is to be reckoned. We shall

employ the symbol h to designate the heat required to raise 1 lb. of water from 32° F. to the temperature t at which steam begins to form. The value of h in thermal units is given, approximately, by the formula

$$h = t - 32 \qquad (3).$$

More exact values, which take account of the variation in the specific heat of water as determined experimentally by Regnault will be found in the last column of Table I. During this first stage, while all the substance still is water, sensibly all the heat that is supplied goes to increase the stock of internal energy which the fluid possesses, for the amount of external work done through the expansion of the water is negligibly small.

58. **Latent Heat of Steam.** During the second stage water at temperature t is changing into steam at temperature t. Much heat is required to produce this change in physical state, although the temperature of the substance does not alter. The heat taken in during this process is called the *latent heat* of steam: in other words, the latent heat of steam is defined as the amount of heat which is absorbed by 1 lb. of water while it changes into 1 lb. of steam under constant pressure, the water having been previously heated up to the temperature at which steam is formed. We shall denote the latent heat by L. The value of L depends on the particular pressure at which the change takes place, Regnault's experiments showing that the latent heat of steam is less at high pressures than at low pressures. A formula for L derived from the results of Regnault's experiments is given in the next paragraph.

Part of the heat taken in during this second stage is spent in doing external work, since the piston rises against the constant pressure of P lbs. per square foot. It is only the remainder of the so-called latent heat L that goes to increase the internal energy of the fluid. The amount spent in doing external work is equal to P multiplied by the change of volume which takes place as the water is converted into steam.

The volume of 1 lb. of water, at such temperatures as are usual in steam-engines, is nearly 0·017 cubic feet. We shall use ω to denote this volume. The external work done during the production of 1 lb. of steam under constant pressure P is therefore

$$\text{External work} = P(V - \omega) \qquad (4).$$

This is the measure of the external work in foot-pounds. It may of course be expressed in thermal units by dividing by J. The external work done in the formation of steam is less at low pressures than at high pressures, and forms a smaller part of the latent heat. Taking the data contained in the table it will be found that when the temperature of formation is 32° Fah. the external work is 54 thermal units, or barely one-twentieth of L: when the temperature is 212° Fah. the external work is 72 thermal units, and at the upper limit of the table it is 86 thermal units.

59. **Total heat of steam.** Adding together the heat taken in during the first and second stages of the imaginary experiment we have a quantity designated by H and called the *total heat* of saturated steam:—

$$H = h + L \quad \text{(5)}.$$

In other words, the total heat of steam is the amount of heat required to raise 1 lb. of water from the standard temperature (32° F.) to the temperature of evaporation and evaporate it there under constant pressure. Regnault's values of H are given in the fourth column of Table I. They are very accurately expressed (in thermal units) by the formula

$$\begin{aligned} H &= 1091{\cdot}7 + 0{\cdot}305\,(t - 32^\circ) \\ &= 1082 + 0{\cdot}305t \quad \text{(6)}. \end{aligned}$$

A similar formula gives approximate values of L, exact enough for use in practical calculations:—

$$L = 1114 - 0{\cdot}7t \quad \text{(7)}.$$

It is, however, generally more convenient to find L from the table, which is readily done, since

$$L = H - h.$$

It follows from these definitions that the whole heat taken in during the formation of 1 lb. of steam, when formed under constant pressure from water at any temperature t_0, is $H - h_0$, where h_0 corresponds to t_0.

To take a numerical example, suppose that steam is formed in a boiler at an absolute pressure of 115 pounds per square inch, the feed-water being supplied at 100° F. Here h_0 is 100 − 32 or 68 thermal units. By the table the temperature of the steam t is

338° F. and H is 1185. The same value of H is obtained by using the formula

$$H = 1082 + 0{\cdot}305 \times 338.$$

Hence the heat taken up by each pound of water in the boiler in first being heated to the boiler temperature and then converted into steam is

$$1185 - 68 \text{ or } 1117 \text{ thermal units.}$$

It is scarcely necessary to add that when steam is condensed under constant pressure an amount of heat equal to L is given out during the change of state from steam to water. Regnault's experiments on the latent heat of steam were in fact made by observing the heat given out when steam from a boiler was led to a calorimeter and was there condensed.

60. Internal energy of steam. Of the whole latent heat of steam L, the part $P(V-\omega)$ is, as has been said above, spent in doing external work. The remainder, namely (in foot-pounds)

$$JL - P(V - \omega),$$

is the increase of internal energy which the substance undergoes during conversion from water at t into steam at t. This quantity, for which it is sometimes convenient to have a separate symbol, will be noted by ρ in thermal units, or $J\rho$ in foot-pounds. In dealing with the heat required to produce steam we adopted the state of water at 32° F. as an arbitrary starting-point from which to reckon the reception of heat. In the same way it is convenient to use this arbitrary starting-point in reckoning what may be called the *internal energy* of the substance, which is the excess of the heat taken in over the external work done by the substance during its reception of heat. Thus the internal energy I of 1 lb. of saturated steam at pressure P is equal to the total heat H, less that part of the total heat which is spent in doing external work, or (in foot-pounds)

$$JI = JH - P(V - \omega),$$

or

$$I = L + h - P(V - \omega)/J = h + \rho \qquad \text{.................(8).}$$

The notion of internal energy is useful in calculating the heat taken in or rejected by steam during any stage of its expansion or compression in an engine. When any working substance passes

from one condition to another, its gain or loss of heat is determined by the equation

Heat taken in = increase of internal energy + external work.

Any of the terms of this equation may be negative; the last term is negative when work is done upon the substance instead of by it.

The relation between the quantities mentioned may be exhibited graphically thus[1]:—

Total heat H

Heat of liquid h — Latent heat of evaporation L

Increase of internal energy during evaporation ρ — External work of evaporation $\frac{P(V-\omega)}{J}$

Internal energy I

61. **Formation of steam otherwise than under constant pressure.** The same equation gives a means of finding the amount of heat required to form steam under any assigned conditions, in place of the condition assumed at the beginning of this chapter, where the formation of steam under constant pressure was considered. Whatever be the condition as to pressure under which the process of formation is carried on, the total heat required is the sum of the internal energy of the steam when formed and the work done by the expanding fluid during the process. Thus in general

$$\text{Heat of formation} = I + \frac{1}{J}\int P dV \;\text{............}\; (9),$$

in thermal units, the limits of integration being the final volume of the steam and the original volume of the water. When saturated steam is formed in a closed vessel of constant volume no external work is done; the heat of formation is then equal to the internal energy I, and is less than the total heat of formation (H) of steam when formed at a constant pressure equal to the pressure finally reached in the vessel, by the quantity $P(V-0{\cdot}017)$.

62. **Wet steam.** In calculations which relate to the action of steam in engines we have generally to deal, not with *dry*

[1] The author is indebted for this suggestion to Prof. Nicolson.

saturated steam, but with *wet* steam, or steam which either carries in suspension, or is otherwise mixed with, a greater or less proportion of water. In every such mixture the steam and water have the same temperature, and the steam is saturated. The *dryness* of wet steam is measured by the proportion q of dry steam in each pound of the mixed substance. When the dryness is known it is easy to determine the other physical constants: thus—

Latent heat of 1 lb. of wet steam $= qL$;

Total heat of 1 lb. of wet steam $= h + qL$;

Volume of 1 lb. of wet steam $= qV + (1 - q)\,0{\cdot}017$

$= qV$ very nearly,

unless the steam is so wet as to consist mainly of water;

Internal energy of 1 lb. of wet steam $= h + q\rho$.

63. **Superheated steam.** Steam is superheated when its temperature is raised, in any manner, above the temperature which corresponds to saturation at the actual pressure. When very highly superheated, steam behaves like a perfect gas, and (to use Rankine's term) may be called *steam gas*. It then follows the equation

$$PV = 85{\cdot}5\tau,$$

and the specific heat at constant pressure, K_p, is about 0·48 thermal unit or 373 foot-pounds. At very low temperatures steam approximates closely to the condition of a perfect gas when slightly superheated, and even when saturated; at high temperatures a much greater amount of superheating is necessary to bring about an approach to the perfectly gaseous state. Rankine has shown that the total heat required for the production of superheated steam under any pressure, when the superheating is so great as to bring the steam to the state of steam gas, may be reckoned by taking the total heat of saturated steam at a low temperature and adding to it the product of K_p into the excess of temperature above that. Treating saturated steam at 32° F. as a gas, he gives the formula

$$H' = H \text{ at } 32^\circ + 0{\cdot}48\,(t' - 32)$$
$$= 1092 + 0{\cdot}48\,(t' - 32)$$

to express the heat of formation of superheated steam, at any temperature t' which is so much above the temperature of

saturation corresponding to the actual pressure that the steam may be treated as a perfect gas. The theoretical basis of this formula will be considered in the next chapter (§ 90). It is not applicable to small amounts of superheating or even to considerable amounts, at such pressures as are usual in steam-engines. A common, but erroneous, practice is to treat the specific heat of steam during superheating under constant pressure as constant (and equal to 0·48), and hence to reckon the total heat by adding to the total heat of the steam when saturated a quantity proportional to the number of degrees of superheating. The experimental data on the subject are still far from complete, but there are sufficient grounds for saying that the quantity of heat taken in per degree during superheating is greater at the initial stage of the process than it is when the amount of superheating has become considerable.

Calculated from its chemical composition, the density of steam gas should be 0·622 times that of air at the same pressure and temperature. The value of γ or K_p/K_v for steam gas is 1·3. These constants dealing as they do with steam which is so highly superheated as to be perfectly gaseous, do not apply to high-pressure steam that is heated but little above its temperature of saturation. The relation of pressure to volume and temperature in the region which lies between the saturated and the perfectly gaseous state has been experimented on by Hirn[1], and formulas which are applicable with more or less accuracy to steam in either the saturated or superheated condition have been devised by Hirn, Zeuner[2], Ritter[3], and others.

64. Isothermal Lines for Steam. The expansion of volume which occurs during the conversion of water into steam under constant pressure—the second stage of the process described in § 53—is isothermal. From what has been already said it is obvious that steam, or any other saturated vapour, can be expanded or compressed isothermally only when wet, and that evaporation (in the one case) or condensation (in the other) must accompany the process. Isothermal lines for a working substance which consists of a liquid and its vapour are straight lines of uniform pressure.

[1] *Théorie Mécanique de la Chaleur.* Part 5, Vol. II.

[2] *Ztschr. d. Vereins deutscher Ingenieure*, vol. xi.

[3] *Wied. Ann.*, 1878. For a discussion of several of these formulas, see a paper by H. Dyer, *Trans. Inst. of Engineers and Shipbuilders in Scotland*, 1885.

65. Adiabatic Lines for Steam. The form of adiabatic lines for substances of the kind just described depends not only on the particular fluid, but also on the proportion of liquid to vapour in the mixture. In the case of steam, it has been shown by Rankine and Clausius that if steam initially dry be allowed to expand adiabatically it becomes wet, and if initially wet (unless very wet[1]) it becomes wetter. To keep steam dry while it expands, doing work, some heat must be supplied during the process of expansion. If the expansion is adiabatic, so that no heat reaches the expanding fluid, a part of the steam is condensed, forming either minute particles of water suspended throughout the mass or a dew upon the surface of the containing vessel. The temperature and pressure fall; and, as that part of the substance which remains uncondensed is saturated, the relation of pressure to temperature throughout the expansion is that which holds for saturated steam. The following formula, a proof of which will be given in the next chapter (§ 81 below), serves to calculate the extent to which condensation takes place during adiabatic expansion, and so allows the relation of pressure to volume to be determined.

Before expansion, let the initial dryness of the steam be q_1 and its absolute temperature τ_1. Then, if it expand adiabatically until its temperature falls to any value τ, its dryness after expansion is

$$q = \frac{\tau}{L}\left(\frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (10).$$

L_1 and L are the latent heats (in thermal units) of 1 lb. of steam before and after expansion respectively. When the steam is dry to begin with, $q_1 = 1$.

This formula, which is applicable with proper values of L to any vapour, may be called the equation of adiabatic expansion or compression. It does not directly give the relation of pressure to volume, but it allows the dryness at any stage of the process to be calculated, and from that (together with the fact that the part which remains in the condition of vapour is saturated) it is easy

[1] When the mixture contains a very large proportion of water to begin with, adiabatic expansion tends to dry it by causing some of the water to evaporate under the reduced pressure which results from the expansion. In the next chapter a graphic method is described of investigating the changes of dryness that are produced by adiabatic expansion, and this may readily be applied to investigate whether the mixture will become drier or wetter in any given case.

to find the volume which the mixture will fill when its pressure has changed to any assigned value. An example may help to make this clear. Suppose for instance that originally dry saturated steam at an absolute pressure of 115·1 pounds per square inch is made to expand adiabatically. Its original volume per lb. (taken from Table I.) is 3·843 cubic feet and its temperature is 338°F. We wish to find the relation of pressure to volume at any stage in the expansion. Take any value of the pressure reached by expansion, say 20·8 pounds per square inch absolute. The corresponding temperature is 230°F. by the table. This gives, for the values of quantities in the adiabatic equation (10),

$$q_1 = 1, \quad \tau_1 = 338 + 461 = 799$$

$$\tau = 230 + 461 = 691,$$

$$L_1 = H_1 - h_1 = 1185{\cdot}0 - 308{\cdot}7 = 876{\cdot}3$$

$$L = H - h = 1152{\cdot}1 - 198{\cdot}7 = 953{\cdot}4.$$

Hence $$q = \frac{691}{953{\cdot}4}\left(\frac{1 \times 876{\cdot}3}{799} + \log_e \frac{799}{691}\right)$$

$$= 0{\cdot}900.$$

This means that by the time the pressure has fallen to 20·8 pounds per sq. inch just one-tenth of the originally dry steam has become condensed into water. The volume occupied by that part of the substance which is still in the state of steam is qV per lb. of the mixture, where V is the volume of 1 lb. of dry steam at the pressure of 20·8 lbs. per sq. inch. Taking the value of V given in the table, namely, 19·18 cubic feet, qV is 17·26 cubic feet. To obtain the whole volume of 1 lb. of the working substance we have in strictness to add to this the volume occupied by that part which has been converted into water, namely, by the fraction of a lb. which is represented by $1 - q$. But this is only 0·1 lb., and its volume is $0{\cdot}1 \times 0{\cdot}017$ or 0·0017 cubic feet—a quantity which is negligible in comparison with the volume occupied by the still uncondensed steam. We conclude that 17·26 cubic feet is the volume of the mixture (per lb.) when its pressure has fallen to 20·8 pounds per square inch by adiabatic expansion; in other words, these numbers determine one point on the adiabatic line which begins with dry steam at a pressure of 115·1 pounds per square inch.

In the same way we may go on to find as many points on an adiabatic line as we please, by taking a series of pressures, each lower than the initial pressure, and finding q for each and from it the volume v, which in ordinary cases is practically equal to qV. We use v here to designate the volume of 1 lb. of the mixture, V being the volume which 1 lb. of saturated steam would occupy at the same pressure and temperature.

The steam may be wet to begin with, and if q_1 have a value much less than unity it will be found on working out examples that q may turn out greater than q_1. This means that in very wet steam adiabatic expansion may reduce the amount of water as the net result of two opposing actions: as the temperature falls during expansion part of the steam initially present becomes condensed; on the other hand, part of the water initially present becomes evaporated because its initial temperature is higher than the temperature which the mixture takes as it expands. With very wet steam the result may be on the whole to make the mixture become drier. An extreme case occurs when all the substance is in the state of water to begin with. Then if adiabatic expansion be allowed to take place steam is formed, and the equation (10) may be applied, by writing $q_1 = 0$, to find how much of the water will be evaporated when the pressure, or the temperature, has fallen to any assigned value.

66. Formula connecting pressure with volume in the adiabatic expansion of steam. Adiabatic curves for steam, whether initially dry or wet, may be calculated in the way that has just been explained, and may then be represented by empirical equations of the form

$$Pv^n = \text{constant},$$

by choosing such values for the index n as will give curves approximating closely to the actual adiabatic curves. A formula of this kind is especially useful for application to cases where the data are the initial pressure and the ratio of expansion r, and it is required to find the pressure after expansion. To find P when the substance has expanded to r times its initial volume,

$$P = \frac{P_1}{r^n} \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\quad (11).$$

The index n has a value which depends on q_1, the initial degree

to fin
has ch
this cle
at an a
expand
Table I.
wish to f
expansion
say 20·8
temperatu
quantities i

Hence q

This means that
pounds per sq. inch
become condensed i
of the substance wh
of the mixture,
pressure of 20
the table, na
obtain the
have in str
whi

ison
We
ture (pe
square in
bers determ
dry steam

ture falls to τ_2. The pressure will then be P_2, namely, the pressure which corresponds in the steam table to τ_2 which is the temperature of the cold body C.

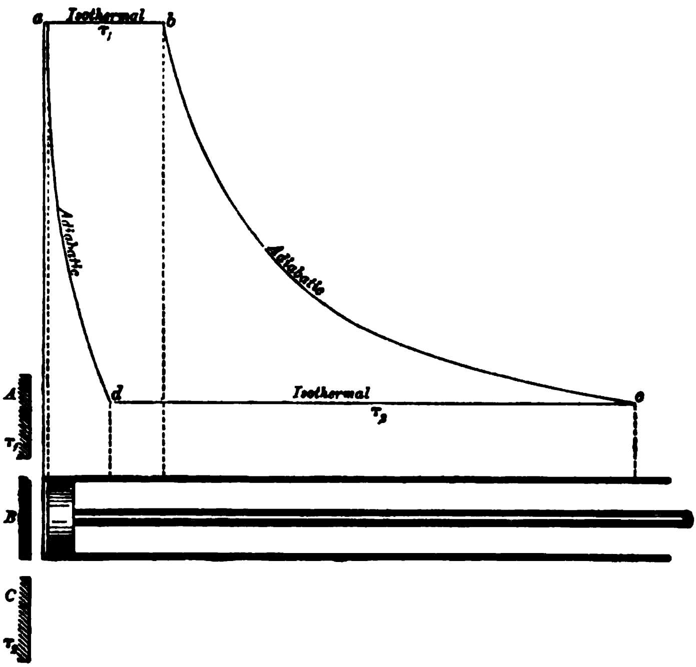


FIG. 14. Carnot's Cycle with water and steam for working substance.

(3) Remove B, apply C, and compress. Steam is condensed jecting heat to C. The action is isothermal, and the pressure ns P_2. Let this be continued until a certain point d is d, after which adiabatic compression will complete the cycle.

Remove C and apply B. Continue the compression, now adiabatic. If the point d has been rightly chosen, complete the cycle by restoring the working fluid to the ter at temperature τ_1.

cator diagram for the cycle is drawn in fig. 14, the having been calculated by the help of the equations , for a particular example, in which $p_1 = 90$ lb. per ($\tau_1 = 781$), and the expansion is continued down to of the atmosphere, 14·7 lb. per square inch ($\tau_2 = 673$).

of dryness of the steam. According to the calculations of Zeuner[1] $n = 1{\cdot}035 + 0{\cdot}1q_1$, so that for

$q_1 = 1$	0·95	0·9	0·85	0·8	0·75	0·7
$n = 1{\cdot}135$	1.130	1·125	1·120	1·115	1·110	1·105.

When it is desired to draw an adiabatic curve for expanding steam, that value of n must be chosen which refers to the degree of dryness at the beginning of the expansion. Rankine gave for this index the value $\frac{10}{9}$, which is too small if the steam be initially dry. It would apply to steam containing about 25 per cent. of water at the beginning of its expansion. We shall see later that the expansion of steam in an actual engine is by no means adiabatic, on account of the transfer of heat which goes on between the working fluid and the metal of the cylinder and piston.

67. Carnot's Cycle with steam for working substance. We are now in a position to study the action of a heat-engine employing water and steam (or any other liquid and its vapour) as the working substance. To simplify the first consideration of the subject as far as possible, let it be supposed that we have, as before, a long cylinder composed of non-conducting material except at the base, and fitted with a non-conducting piston; also a source of heat A at some temperature τ_1; a receiver of heat, or as we may now call it, a condenser, C, at some lower temperature τ_2; and also a non-conducting cover B (as in § 41). Then Carnot's cycle of operations can be performed as follows. To fix the ideas, suppose that there is 1 lb. of water in the cylinder to begin with, at the temperature τ_1:—

(1) Apply A, and allow the piston to rise under the constant pressure P_1 which corresponds to the temperature τ_1. The water will take in heat and be converted into steam, expanding isothermally at the temperature τ_1. This part of the operation is shown by the line ab in fig. 14.

(2) Remove A and apply B. Allow the expansion to continue adiabatically (bc), with falling pressure, until the tempera-

[1] *Grundzüge der Mech. Wärmetheorie*, p. 342, or *Technische Thermodynamik*, Vol. II. (1890) p. 74. See also Grashof, *Resultate aus der Mech. Wärmetheorie*, § 37. In the adiabatic compression of wet steam $n = 1{\cdot}034 + 0{\cdot}11q_1$, where q_1 is the dryness at the beginning of compression.

ture falls to τ_2. The pressure will then be P_2, namely, the pressure which corresponds in the steam table to τ_2 which is the temperature of the cold body C.

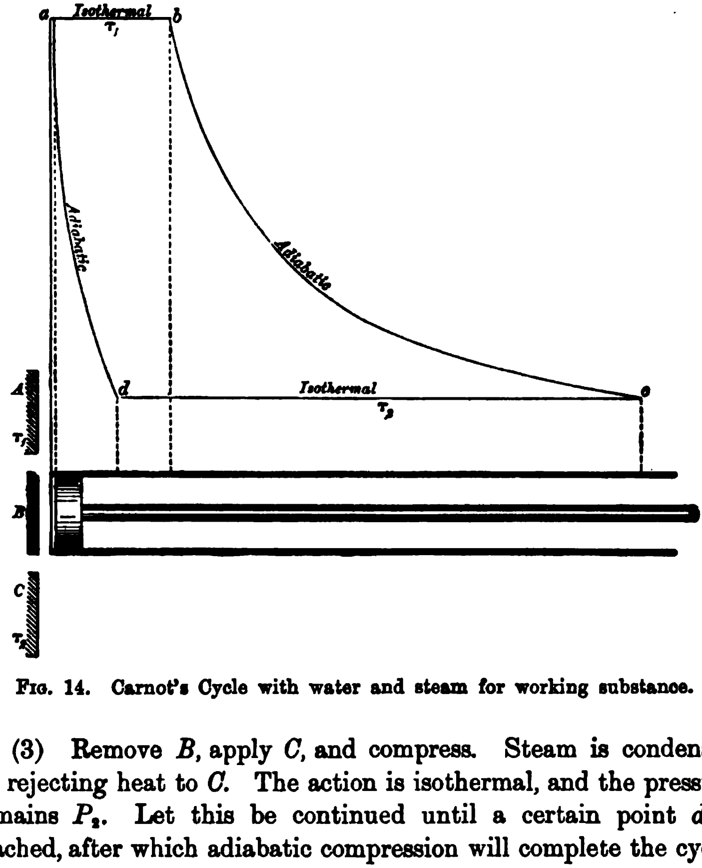


FIG. 14. Carnot's Cycle with water and steam for working substance.

(3) Remove B, apply C, and compress. Steam is condensed by rejecting heat to C. The action is isothermal, and the pressure remains P_2. Let this be continued until a certain point d is reached, after which adiabatic compression will complete the cycle.

(4) Remove C and apply B. Continue the compression, which is now adiabatic. If the point d has been rightly chosen, this will complete the cycle by restoring the working fluid to the state of water at temperature τ_1.

The indicator diagram for the cycle is drawn in fig. 14, the lines bc and da having been calculated by the help of the equations in §§ 65 and 66, for a particular example, in which $p_1 = 90$ lb. per square inch ($\tau_1 = 781$), and the expansion is continued down to the pressure of the atmosphere, 14·7 lb. per square inch ($\tau_2 = 673$).

Since the process is reversible, and since heat is taken in only at τ_1 and rejected only at τ_2, the efficiency is

$$\frac{\tau_1 - \tau_2}{\tau_1}.$$

The heat taken in per lb. of the fluid is L_1, and the work done is

$$\frac{L_1(\tau_1 - \tau_2)}{\tau_1},$$

a result which may be used to check the calculation of the lines in the diagram by comparing it with the area which they enclose. It will be seen that the whole operation is strictly reversible in the thermodynamic sense.

Instead of supposing the working substance to consist wholly of water at a and wholly of steam at b, the operation ab might be taken to represent the partial evaporation of what was originally a mixture of steam and water. The heat taken in would then be $(q_b - q_a)\, L$, and as the cycle would still be reversible the area of the diagram would be

$$\frac{L\,(q_b - q_a)\,(\tau_1 - \tau_2)}{\tau_1}.$$

68. Efficiency of a perfect Steam-engine. Limits of temperature. If the action here described could be realised in practice, we should have a thermodynamically perfect steam-engine using saturated steam. Like any other perfect heat-engine an ideal engine of this kind has an efficiency which depends upon the temperatures between which it works, and upon nothing else. The fraction of the heat supplied to it which such an engine would convert into work would depend simply on the two temperatures, and therefore on the pressures, at which the steam was produced and condensed respectively.

It is interesting therefore to consider what are the limits of temperature between which steam-engines may be made to work. The temperature of condensation is limited by the consideration that there must be an abundant supply of some substance to absorb the rejected heat; water is actually used for this purpose, so that τ_2 has for its lower limit the temperature of the available water-supply.

To the higher temperature τ_1 and pressure P_1 a practical limit is set by the mechanical difficulties, with regard to strength and

to lubrication, which attend the use of high-pressure steam. By a very special construction of engine and boiler Mr L. Perkins has been able to use steam with a pressure as high as 500 lbs. per square inch; with engines of the usual construction the pressure ranges from about 200 lbs. downwards.

This means that the upper limit of temperature, so far as the steam is concerned, is barely 400° F. A steam-engine, therefore, under the most favourable conditions, comes very far short of taking full advantage of the high temperature at which heat is produced in the combustion of coal. From the thermodynamic point of view the worst thing about a steam-engine is the irreversible drop of temperature between the furnace and the boiler. The combustion of the fuel supplies heat at a high temperature: but a great part of the convertibility of that heat into work is at once sacrificed by the fall in temperature which is allowed to take place before the conversion into work begins.

If the temperature of condensation be taken as 60° F., as a lower limit, the efficiency of a perfect steam-engine, using saturated steam, would depend on the value of P_1, the absolute pressure of production of the steam, as follows:—

Perfect steam-engine, with condensation at 60° F.,

P_1 in lbs. per square inch being	40	80	120	160	200
Highest ideal efficiency	= ·284	·326	·350	·368	·381

But it must not be supposed that these values of the efficiency are actually attained, or are even attainable. Many causes conspire to prevent steam-engines from being thermodynamically perfect, and some of the causes of imperfection cannot be removed. These numbers will serve, however, as one standard of comparison in judging of the performance of actual engines, and as setting forth the advantage of high-pressure steam from the thermodynamic point of view.

69. Efficiency of an engine using steam non-expansively. As a contrast to the ideally perfect steam-engine of § 67 we may next consider a cyclic action such as occurred in the early engines of Newcomen or Leupold, when steam was used non-expansively,—or rather, such an action as would have occurred in engines of this type had the cylinder been a perfect non-conductor of heat. In that case the volume of steam formed is equal to the

volume swept through by the piston. We may represent the action of such an engine thus:

(1) Apply the hot body A and evaporate the water as before at P_1. Heat taken in, per lb. of the working fluid, $= L_1$.

(2) Remove A and apply the cold body C. This at once condenses a part of the steam, and reduces the pressure to P_2.

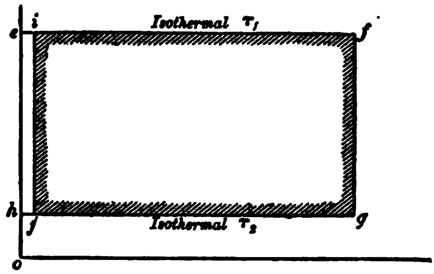


FIG. 15.

(3) Compress at P_2, in contact with C, till condensation is complete, and water at τ_2 is left.

(4) Remove B and apply A. This heats the water again to τ_1 and completes the cycle. Heat taken in $= h_1 - h_2$.

The indicator diagram for this series of operations is shown in fig. 15, where $oe = P_1$ and $oh = P_2$.

Here the action is not reversible. To calculate the efficiency

$$\frac{\text{Work done}}{\text{Heat taken in}} = \frac{(P_1 - P_2)(V_1 - \omega)}{J(L_1 + h_1 - h_2)}.$$

The values of this will be found to range from 0·067 to 0·072 for the values of P_1 which are stated in § 68, when the temperature of condensation is 60° F. Contrast these numbers with the much higher efficiencies found in the last paragraph for a perfect steam-engine, following Carnot's cycle.

The efficiency of the actual Newcomen engine was much lower even than this calculation indicates, because in every stroke of the piston a large part of the steam entering the cylinder was at once condensed upon the sides, and the volume of steam which had to be supplied from the boiler was therefore much greater than the volume swept through by the piston.

70. Engine with separate organs. In the ideal engine represented in fig. 14 the functions of boiler, cylinder, and

condenser are combined in a single vessel; but after what has been said in Chapter II. it is scarcely necessary to remark that, provided the working substance passes through the same cycle of operations it is indifferent whether these are performed in several vessels or

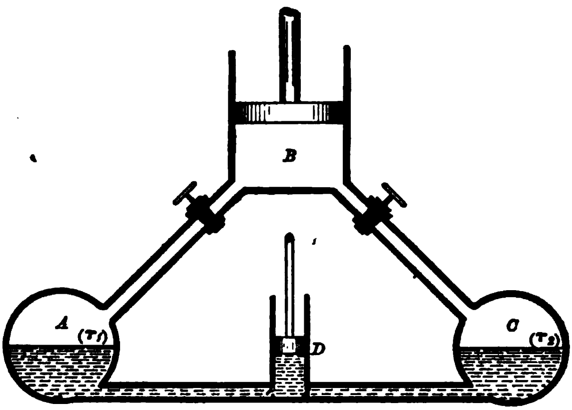


FIG. 16. Organs of a Steam-Engine.

in one. To approach a little more closely the conditions which hold in practice, we may think of the engine which performs the cycle of § 69 as consisting of a boiler A (fig. 16) kept at τ_1, a non-conducting cylinder and piston B, a surface condenser C kept at τ_2, and a feed-pump D which restores the condensed water to the boiler. Then for every pound of steam supplied and used non-expansively as in § 69, we have

$$\text{work done on the piston} = (P_1 - P_2)\, V_1;$$

but the amount of work which has to be expended in driving the feed-pump is $(P_1 - P_2)\,\omega$. Deducting this, the net work done per lb. of steam is the same as before, and the heat taken in is also the same. An indicator diagram taken from the cylinder would give the area $efgh$ (fig. 15), where

$$oe = P_1,\ ef = V_1,\ oh = P_2;$$

an indicator diagram taken from the pump would give the negative area $hjie$, where ei is the volume of the feed-water, or 0·017 cub. ft. The difference between these two areas, namely, the area $ifgh$ which is shaded in the figure, is the diagram of the complete cycle gone through by each pound of the working substance. In experimental measurements of the work done in steam-engines, only the action which occurs within the cylinder is shown on the

indicator diagram. From this the work spent on the feed-pump is to be subtracted if we wish to make a rigorous determination of the thermodynamic efficiency. If the feed-water be at any temperature τ_0 other than the temperature of condensation τ_2, it is clear that the heat taken in is $H_1 - h_0$ instead of $H_1 - h_2$.

71. How nearly may the process in a Steam-engine be reversible? We have now to inquire how nearly, with the engine of fig. 16 (that is to say, with an engine in which the boiler and condenser are separate from the cylinder), we can approach the reversible cycle of § 67. The first stage of that cycle corresponds to the *admission* of steam from the boiler into the cylinder, for during admission of steam to the cylinder a corresponding quantity of steam is being formed in the boiler. Then the point known as the point of *cut-off* is reached, at which admission ceases, and the steam already in the cylinder is allowed to expand, exerting a diminishing pressure on the piston. This is the second stage, or the stage of *expansion*. The process of expansion may be carried on until the pressure falls to that of the condenser, in which case the expansion is said to be complete. At the end of the expansion *release* takes place, that is to say, communication is opened with the condenser. Then the return stroke begins, and a period termed the *exhaust* occurs, that is to say, steam passes out of the cylinder, into the condenser, where it is condensed at pressure P_2, which is felt as a *back pressure* opposing the return of the piston. So far, all has been essentially reversible, and identical with the corresponding parts of Carnot's cycle.

But we cannot complete the cycle as Carnot's cycle was completed. The existence of a separate condenser makes the fourth stage, that of adiabatic compression, impracticable, and the best we can do is to continue the exhaust until condensation is complete, and then return the condensed water to the boiler by means of the feed-pump.

It is true that we may, and in actual practice do, stop the exhaust before the return stroke is complete, and compress that portion of the steam which remains below the piston, but this does not materially affect the thermodynamic efficiency; it is done partly for mechanical reasons, and partly to avoid loss of power through clearance (see Chap. V.). In the present instance it is

supposed that there is no clearance, in which case this compression is out of the question. The indicator diagram given by a cylinder in which steam goes through the action described above is drawn to scale in fig. 17 for a particular example, in which it is supposed that dry saturated steam is admitted to the cylinder at an absolute pressure of 90 lbs. per square inch, and is then expanded adiabatically to twelve times its original volume. This brings it down to

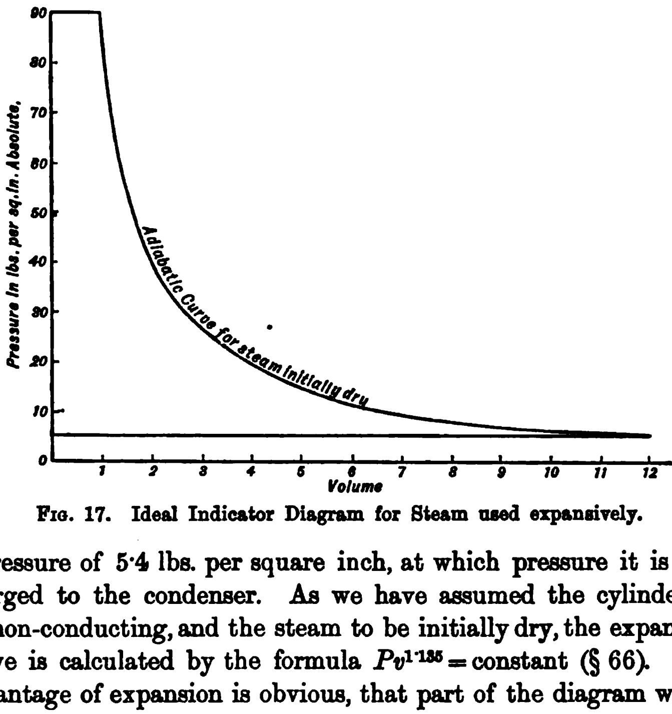


FIG. 17. Ideal Indicator Diagram for Steam used expansively.

a pressure of 5·4 lbs. per square inch, at which pressure it is discharged to the condenser. As we have assumed the cylinder to be non-conducting, and the steam to be initially dry, the expansion curve is calculated by the formula $Pv^{1 \cdot 135} = \text{constant}$ (§ 66). The advantage of expansion is obvious, that part of the diagram which lies under the curve being so much clear gain, as compared with the case dealt with in § 69.

To calculate the performance, we have

Work done per lb. during admission $= P_1 V_1$;

" " during expansion to volume $r V_1 = \dfrac{P_1 V_1 - P_2 r V_1}{n-1}$

(by § 37), $= \dfrac{P_1 V_1 - P_2 r V_1}{0 \cdot 135}$;

Work spent during return stroke $= P_2 r V_1$;

" " on the feed-pump $= (P_1 - P_2)\, 0 \cdot 017$;

Heat taken in $= H_1 - h_0$.

Then, by comparing the net amount of work done with the heat taken in we may find the efficiency. Another method of calculating the work done in this cycle of operations will be given in the next chapter.

In the above example the expansion is complete, that is to say, the substance is allowed to expand until its temperature falls to that of the condenser or cold body into which heat is to be rejected.

When the expansion is incomplete, as it generally is in practice, the expression given above for the work done during expansion still applies if we understand P_2 to be the pressure at the end of expansion, while the work spent on the steam during the back-stroke is $P_b r V_1$ and that spent on the feed-pump is $(P_1 - P_b)\, 0{\cdot}017$, P_b being the back-pressure. Incomplete expansion is illustrated by fig. 18, where the steam is supposed to escape after expanding

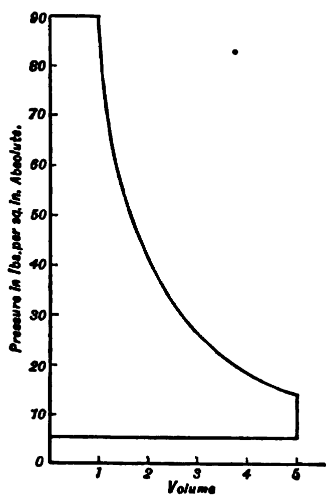


FIG. 18. Incomplete expansion.

to five times its initial volume. It results simply in a loss of the work which is represented by the toe of the diagram, that is to say, by the difference of areas between this and the last figure.

It is easy to extend these calculations to cases where the steam, instead of being initially dry, is supposed to have any assigned degree of wetness. The efficiency which is calculated in this way, which for the present purpose may be called the

theoretical efficiency corresponding to the assumed conditions of working, is always less than the ideal highest efficiency of a perfect engine working between the same limits of temperature. This is because of the absence of the compression which formed the fourth stage in Carnot's cycle, and had the effect of bringing the temperature up to the top of the range before the substance began to take in heat. Without compression some of the heat is taken in at temperatures below the highest temperature τ_1, and any heat taken in at a lower temperature cannot contribute so much work as if it had been taken in at τ_1. But even the theoretical efficiency working in this way without compression, short as it falls of the ideal of a perfect engine, is considerably greater than can be realized in practice when the same boiler and condenser temperatures are used, and the same ratio of expansion. The reasons for this will be considered in Chapter V.; at present the fact is mentioned to guard the reader from supposing that the results which the above formulas give apply to actual engines.

72. Engine in which the steam is kept dry during expansion. Another case of theoretical interest is presented if we suppose the steam instead of expanding adiabatically to be kept dry and saturated during expansion, enough heat being communicated to it from a steam jacket to keep it dry. In the real use of a steam jacket the steam in the cylinder is not as a rule kept quite dry. A jacket which would keep the steam dry throughout expansion would be giving as much heat as a jacket can in the extreme case be expected to give. The jacket gives heat by evaporating the water which the working fluid contains: if that were all evaporated there would be practically no further communication of heat since the superheating of a dry gas by conduction is slow. The limiting case in the action of a jacket may be said to be reached when the steam is kept dry during expansion. In that case the expansion curve has (by § 56) the equation

$$PV^{\frac{17}{16}} = P_1V_1^{\frac{17}{16}}.$$

The work done during expansion from V_1 to V_2 is

$$\frac{P_1V_1 - P_2V_2}{n-1} = 16(P_1V_1 - P_2V_2).$$

The work done during admission is P_1V_1 and the work spent

during exhaust is P_2V_2. Hence the net amount of work done by the steam is $17\,(P_1V_1 - P_2V_2)$.

The heat received from the boiler is $H_1 - h_2$; the heat rejected to the condenser is $H_2 - h_2$. Hence the quantity of heat H_j received from the jacket is given by the equation

$$H_1 - H_2 + H_j = 17\,(P_1V_1 - P_2V_2),$$

or
$$H_j = 17\,(P_1V_1 - P_2V_2) - 0{\cdot}305\,(t_1 - t_2) \times 778.$$

The efficiency is $$\frac{17\,(P_1V_1 - P_2V_2)}{H_1 - h_2 + H_j}.$$

Another method will be given in Chapter IV. of calculating the heat supplied by the jacket in this limiting case of expansion along the saturation curve.

CHAPTER IV.

FURTHER POINTS IN THE THEORY OF HEAT-ENGINES.

73. Rankine's statement of the Second Law. Rankine, to whom with Clausius and Lord Kelvin is due the development of the theory of heat-engines from the point at which it was left by the "Réflexions" of Carnot and the experiments of Joule, has, in his "Manual of the Steam-Engine and other Prime Movers," stated the second law of thermodynamics in a form which is neither easy to understand, nor obvious, as an experimental result, when understood. His statement runs:—

"If the absolute temperature of any uniformly hot substance be divided into any number of equal parts, the effects of those parts in causing work to be performed are equal."

To make this intelligible we may suppose that any quantity q of heat from a source at temperature τ_1 is taken by the first of a series of perfect heat-engines, and that this engine rejects heat at a temperature τ_2 which is less than τ_1 by a certain interval $\Delta\tau$. Let the heat so rejected by the first engine form the heat supply of a second perfect engine working from τ_2 to τ_3 through an equal interval $\Delta\tau$; let the heat which it in turn rejects form the heat-supply of a third perfect engine working again through an equal interval from τ_3 to τ_4; and so on. The efficiencies of the several engines are (by § 47)

$$\frac{\Delta\tau}{\tau_1}, \quad \frac{\Delta\tau}{\tau_2}, \quad \frac{\Delta\tau}{\tau_3}, \text{ \&c.}$$

The amounts of heat supplied to them are

$$q, \, q\frac{\tau_2}{\tau_1}, \quad q\frac{\tau_3}{\tau_1}, \text{ \&c.}$$

Hence the amount of work done by each engine is the same, namely,

$$q \frac{\Delta \tau}{\tau_1}.$$

Thus Rankine's statement is to be understood as meaning that each of the equal intervals into which any range of temperature may be divided is equally effective in allowing work to be produced from heat when heat is made to pass, doing work in the most efficient possible way, through all the intervals from the top to the bottom of the range.

74. Absolute Temperature: Lord Kelvin's scale. In the preceding chapters we have been using the imaginary perfect gas thermometer as the means of framing a scale of temperatures. In other words, our scale has been such that equal intervals of temperature are defined as those which correspond to equal amounts of expansion of a perfect gas under constant pressure. We have defined τ by means of the formula $V = c\tau$, P being constant. And seeing that air behaves as a nearly perfect gas this scale is practically realised by the air thermometer.

Starting from this definition of temperature we have found by an application of Carnot's principle that a reversible engine working between a hot source A and cold receiver of heat C takes in from the source and gives out to the receiver quantities of heat Q_A and Q_C which are proportional to the absolute temperatures of the source and receiver respectively, as defined by reference to the perfect gas thermometer.

Hence we might have defined temperature in a very different way and still have arrived at just the same scale. We might have said, let the temperatures of A and C be specified by two numbers which shall be proportional to the heat taken in and given out respectively by a reversible heat-engine when working with A for source and C for receiver of heat. This method of defining absolute temperature was proposed by Lord Kelvin. It gives a scale which is truly absolute in the sense of being independent of the properties of any gas or other substance, real or imaginary. The scale so obtained coincides with the scale of the perfect gas thermometer.

Lord Kelvin's method of devising a scale of absolute temperatures may also be put in a somewhat different fashion, thus:— Starting with any arbitrary temperature let a series of intervals be taken such that equal amounts of work will be done by every

one of a series of reversible engines, each working with one of these intervals for its range and each handing on to the engine below it the heat which it rejects (so that the heat rejected by the first forms the supply of the second, and so on). Then call these intervals equal. This is only another way of putting the definition of absolute temperature which has just been quoted: it is suggested by what has been said in the last paragraph about Rankine's statement of the Second Law.

The scale of the actual air thermometer would be in perfect agreement with Lord Kelvin's absolute scale if the laws stated in Chapter II. were rigorously true of air, namely, Regnault's law, according to which the specific heat at constant pressure is constant (§ 33), and Joule's law, according to which there is no change of temperature when a gas expands without doing external work and without receiving or rejecting heat (§ 34). The experiments by which Joule established his law have been already described. Reference has also been made to the subsequent experiments of a more searching kind, devised by Lord Kelvin, and carried out by him in conjunction with Joule, in which air was forced slowly through a porous plug to see whether its temperature became changed, which have shown that air does not conform with perfect exactness to Joule's law[1]; but the deviations are so slight that for all practical purposes the scale of the air thermometer may be taken as agreeing with the absolute scale[2].

Actual air thermometers may be made for use in two ways: In one the pressure is kept constant and the volume is allowed to expand or contract as the temperature varies; in the other the volume is kept constant by adapting the pressure to the temperature which is being measured, and the temperature is then taken to be proportional to the pressure. This latter is the more practicable form: it is called the constant volume air thermometer. The air must be perfectly dry: if there is any water vapour in it the volume in the one case or the pressure in the other may be far from proportional to the temperature.

75. Calculation of the Density of Saturated Steam. In our account of the physical properties of saturated steam it

[1] See Lord Kelvin's *Collected Papers*, Vol. I. p. 333.

[2] For a comparison by Rowland based on the experiments of Joule and Lord Kelvin, see *Proceedings of the American Academy*, 1879, also Professor Peabody's *Thermodynamics of the Steam-Engine*, Chapter VI.

was mentioned that the volumes of 1 lb. stated in the third column of Table I. were not found by direct experiment, but were calculated from other known properties. To explain how this is done we may revert to the ideally perfect steam-engine of § 67, in which Carnot's cycle is followed with water and steam for working substance. We saw that this gave an indicator diagram (fig. 14) with two lines of uniform pressure (isothermals) connected by two adiabatic curves. The heat taken in was L per lb. of working substance, and since the engine was reversible its efficiency was

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

from which it followed that the work done, or the area of the diagram, was

$$\frac{L(\tau_1 - \tau_2)}{\tau_1}.$$

This is in thermal units: to reduce it to foot-pounds we multiply by J. Now suppose that the engine works between two temperatures which differ by only a very small amount. We may call the temperatures τ and $\tau - \delta\tau$, $\delta\tau$ being the small interval through which the engine works. The above expression for the work done becomes (in foot-pounds)

$$\frac{JL\delta\tau}{\tau}.$$

The indicator diagram is now a long narrow strip (fig. 19). Its length ab is $V - \omega$, V being the volume of 1 lb. of steam and ω the volume of 1 lb. of water, or say 0·017 cubic feet. Its height is δP, where δP is the difference between the pressure in ab and that in cd. In other words, since the steam is saturated in cd as well as in ab, δP is the difference in the pressure of saturated steam due to the difference in temperature $\delta\tau$. When δP is made very small, the area of the diagram becomes more and more nearly equal to the product of the length by the height, namely, $\delta P(V - \omega)$. This is equal to the work done, whence

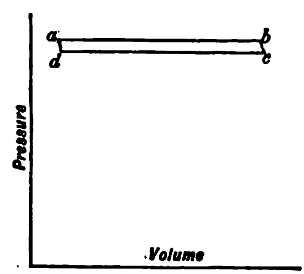


FIG. 19.

$$\delta P(V - \omega) = \frac{JL\delta\tau}{\tau} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1).$$

This equation is only approximate when the interval $\delta\tau$ (or δP) is a small finite interval. In the limit, when the interval is made indefinitely small, it becomes exact and may then be written

$$V - \omega = \frac{JL}{\tau}\frac{d\tau}{dP} \quad \text{.......................(2)},$$

$\frac{d\tau}{dP}$ being the rate at which the temperature of saturated steam alters relatively to the pressure when the temperature is τ.

Thus we have the equation

$$V = \omega + \frac{JL}{\tau}\frac{d\tau}{dP}$$

as a means of calculating the volume of 1 lb. of steam when the values of L and of $\frac{d\tau}{dP}$ for various temperatures are known. Regnault's experiments have determined L, and by giving the relation of P to τ they have also given data from which it is easy to find $\frac{d\tau}{dP}$ either by measuring the slope of a tangent to the curve of τ and P or by differentiating a formula such as equation (1) of § 55 which expresses the experimental relation between these two quantities. It is in this way that the values of V in the Table have been determined.

The advantage of the method is that L and $\frac{d\tau}{dP}$ can be measured more accurately than V could itself be measured, and thus the values of V obtained indirectly from them are more likely to be right than those obtained by direct experiment. The formula shows that the numbers found in this way depend on J, and are nearly proportional to it, since ω is small; and hence, as has been remarked before, the numbers given in the third column of the table have required alteration from those given by Rankine, because 778 is accepted as the mechanical equivalent of heat instead of 772.

76. Extension of the above result to other changes of physical state. In equation (2), above, the left-hand side is positive, since V the volume of 1 lb. of steam is greater than ω the volume of 1 lb. of water. The right-hand side must also be positive, and hence it is that $\frac{d\tau}{dP}$ is positive, or

in other words, that increasing the pressure under which steam is formed raises the boiling point. The equation might evidently be applied in the reverse way to that indicated above (for finding V); in other words, if the amount by which the volume increases when water changes into steam were given we might employ that to calculate $\frac{d\tau}{dP}$, the rate at which the boiling point is raised by increase of pressure.

Further, the reasoning by which this equation was arrived at was perfectly general and was in no way restricted to the case of steam. The engine whose indicator diagram is sketched in fig. 19 might have anything for working substance, the isothermal line of the first operation, during which heat is taken in, representing in the most general way the change of volume which occurs while any working substance changes its physical state. In the example already dealt with the change is from liquid to vapour. But we might begin with a solid substance previously raised to the temperature τ at which it begins to melt and let the first stage in the cycle consist in the expansion of the substance while it passes from the solid to the liquid state, the substance doing external work by overcoming a constant pressure as it expands. All the steps in the argument remain unaffected, and hence the equation may be written thus with reference to any transformation of state on the part of any substance,

$$U - U' = \frac{J\lambda}{\tau}\frac{d\tau}{dP} \quad\text{..........................(3)},$$

where U' is the volume of unit mass of the substance in the original state, U is the volume after the transformation has taken place, λ is the heat absorbed while the transformation is going on (the latent heat of fusion or of evaporation as the case may be), and $\frac{d\tau}{dP}$ is the rate at which the temperature of the transformation (say the melting-point or the boiling-point) is affected by altering the pressure under which the change of state occurs.

If a solid body expands on melting, U is greater than U', and consequently $\frac{d\tau}{dP}$ must be positive: in other words, the melting-point will in that case be raised by applying pressure.

On the other hand if the substance contracts on melting,

$U - U'$ is negative and τ must then *decrease* relatively to P, that is to say, the melting-point is then lowered by applying pressure. This is the case with ice. From the known amount by which ice contracts when it melts James Thomson (in 1849) first applied this method of reasoning to show that the melting-point of ice must be lowered to a definite extent when the ice is melted under any assigned pressure, and the result was afterwards verified by an experiment of his brother, Lord Kelvin. The amount by which the melting-point is lowered is about 0·0135° F. for each atmosphere of pressure[1].

77. Drying of steam by throttling or wire-drawing. When dry steam expands without doing work and without receiving or rejecting heat it becomes superheated; and if wet to begin with it becomes drier. This is because the total heat of steam (H) is less at low pressure than at high. Suppose for instance that steam is flowing through a small pipe or orifice from a chamber where the pressure is P_1 to another where it is P_2. Such an action happens in steam-engines in the movement of steam through contracted pipes and passages between the boiler and the valve chest: the steam becomes reduced in pressure and is said to be throttled or "wire-drawn." Eddies are formed in rushing through the constricted openings and the energy expended in forming them is frittered down into heat as the eddies subside. To calculate the amount of drying, in the case of steam that is initially wet, we have (if no heat enters or leaves the fluid)

$$q_1L_1 + h_1 = q_2L_2 + h_2,$$

where the suffixes 1 and 2 refer to the condition before and after throttling respectively. It is assumed that a steady condition exists before and also after the throttling and that the chambers are large, so that the stream of steam has no kinetic energy worth

[1] See Lord Kelvin's *Collected Papers*, Vol. I. p. 156 and p. 165. The numerical result stated in the text is obtained as follows:—A pound of water changes its volume in freezing from 0·016 to 0·0174 cub. ft., and gives out 142 units of heat. Hence

$$\frac{d\tau}{dP} = \frac{0·0014 \times 493}{142 \times 778} = 0·0000065,$$

and if δP be one atmosphere or (say) 2160 lbs. per sq. ft., $\delta\tau$ is $2160 \times 0·0000065$ or 0·0135° Fah.

taking account of either before it passes the orifice or after it has passed and the eddies have subsided. From this

$$q_2 = \frac{q_1 L_1 + h_1 - h_2}{L_2}.$$

In the same way dry steam escaping at high pressure from a boiler into the atmosphere is superheated at a little distance from the orifice; further off it becomes condensed by loss of heat to the air.

78. Engine receiving heat at various temperatures. In Carnot's cycle it was assumed that the working substance took all its heat in at the higher limit of temperature τ_1. Important cases arise in which heat is taken in partly at one and partly at other temperatures in a single cycle of operations. With regard to every such quantity of heat the result still applies that the greatest fraction that can be converted into work under ideally favourable conditions is represented by the difference between its temperatures of reception and rejection, divided by the absolute temperature of reception.

Thus if Q_1 represents that part of the whole heat which is taken in at τ_1, and Q_2 represents what is taken in at some other temperature τ_2, Q_3 at τ_3 and so on, and if τ_0 be the temperature at which the engine rejects heat, the whole work done, if the processes within the engine are reversible,

$$W = \frac{Q_1(\tau_1 - \tau_0)}{\tau_1} + \frac{Q_2(\tau_2 - \tau_0)}{\tau_2} + \frac{Q_3(\tau_3 - \tau_0)}{\tau_3} + \dots \text{ etc. } \dots (4).$$

It is perhaps worth while to point out the analogy here to the supposititious case of a water-wheel working by gravity and receiving water into its buckets at different heights above the level at which water is discharged from them. Let M_1, M_2 and so on be the quantities of water received at heights l_1, l_2 etc. above any datum level, and let l_0 be the height above the same datum level at which the water leaves the wheel. If the wheel is perfectly efficient (and here again the test of perfect efficiency is reversibility) the work done is

$$M_1(l_1 - l_0) + M_2(l_2 - l_0) + M_3(l_3 - l_0) + \dots \text{ etc.}$$

Comparing the two cases we see that the quantity $\frac{Q_1}{\tau_1}$ is the analogue in the heat-engine of M_1 in the water-wheel and so on.

The amount of work which can be got out of a given quantity of heat by letting it down to an assigned level of temperature is not simply proportional to the product of the quantity of heat by the fall of temperature, but to the product of $\frac{Q}{\tau}$ by the fall of temperature. On the strength of this analogy Zeuner has called the quantity $\frac{Q}{\tau}$ the "heat weight" of a quantity of heat Q obtainable at a temperature τ.

Another way of expressing the matter has a wider application. Let the engine as before take in quantities of heat represented by Q_1, Q_2, Q_3 etc. at τ_1, τ_2, τ_3 and let $-Q_0$ represent the heat rejected at τ_0, the negative sign being used to distinguish heat rejected from heat received. Then by the principle that in a reversible cycle the heat rejected is to the heat taken in as the absolute temperature of rejection is to the absolute temperature of reception, we have

$$\frac{-Q_0}{\tau_0} = \frac{Q_1}{\tau_1} + \frac{Q_2}{\tau_2} + \frac{Q_3}{\tau_3} + \ldots,$$

from which

$$\Sigma \frac{Q}{\tau} = 0 \ldots\ldots\ldots\ldots\ldots\ldots(5),$$

when the summation is effected all round the reversible cycle. It is clear that this result may be at once extended to cases where heat is given out at various temperatures as well as taken in at various temperatures, Q being taken positive or negative according as heat is being received or rejected.

In cases where changes of temperature are going on continuously while heat is being taken in or given out, we cannot divide the reception or rejection of heat into a limited number of steps, as has been done above. But the equation may be adapted to this most general case by writing it

$$\int \frac{dQ}{\tau} = 0 \ldots\ldots\ldots\ldots\ldots\ldots(6),$$

integration being performed round the whole cycle.

79. Application to the case of a steam-engine working without compression, but with complete adiabatic expansion. In § 71 we considered the action of an ideal steam-engine in which the steam formed at τ_1 was expanded adiabatically and

fully, that is to say, down to the pressure corresponding to the temperature of the condenser τ_2 and was there condensed, the condensed water being then restored to the boiler by a feed-pump and thus heated again to τ_1 to complete the cycle. This cycle is specially important in the discussion of steam-engines because it represents the ideally best performance of an engine which uses a feed-pump to return the condensed water directly from the condenser to the boiler, namely, the performance which such an engine might achieve provided the expansion were complete, so that there should be no sudden drop of pressure at release, and provided the cylinder and piston were perfect non-conductors. The efficiency in this cycle falls short of the Carnot limit

$$\frac{\tau_1 - \tau_2}{\tau_1}$$

because in the fourth stage of the cycle the working substance has its temperature raised from τ_2 to τ_1, not by adiabatic compression, as in Carnot's cycle (§ 67), but by being brought into contact with the contents of the boiler, which are kept at τ_1. Consequently heat enters it in this stage by a non-reversible process: in all other respects however the cycle is reversible.

But we may regard this as a strictly reversible cycle if we think of the feed-water as taking up its heat by infinitesimal instalments at a series of temperatures ranging from τ_2 up to τ_1 from a series of imaginary sources each of which has the same temperature as the water when the water is brought into contact with it. One may realise this notion by thinking of the feed-pipe as passing through a heated channel the temperature of which is τ_1 close to the boiler and tapers down to τ_2 close to the condenser. Thus the feed-water would have its temperature raised gradually and would nowhere be brought into contact with a source at a temperature different from the temperature which it had itself then reached. With such an arrangement as this it is clear that the engine becomes a strictly reversible engine, receiving portions of its heat, however, at various temperatures. But the action of the engine is in no way altered by this imaginary arrangement of the feed-pipe, nor is the total supply of heat in any way altered. The notion of gradual heating in the feed-pipe has been introduced merely to show that the cycle is a reversible cycle if we take account of the fact that heat is received not all at the top of the range of temperature, but partly at lower

temperatures. Every part of the heat which the substance receives is used in the most efficient possible way, *after it has been taken in,* so that the expression

$$\frac{\tau - \tau_2}{\tau}$$

measures the efficiency of the transformation into work of each portion of the heat, τ being the particular temperature at which the working substance happens to be when it takes in that portion of the heat. The only non-reversible feature in the action of this engine is the flow of heat from a source at τ_1 into the feed-water while the temperature of the feed-water is less than τ_1; and we get rid of this partial non-reversibility by taking as the temperature of reception of each portion of the heat that temperature which the working substance has when the portion in question was taken in. It will be evident that these remarks are of general application, and that when this understanding is accepted, both with regard to the temperatures of reception and rejection of heat, the process in any heat-engine is to be taken as reversible provided the expansions and compressions which occur in it are themselves reversible in the sense which has been explained in § 50. With a source of heat at a given temperature the heat can be turned to account most efficiently only when all the heat is taken in while the working substance is at that temperature, and it is only then that the greatest value of the efficiency, namely, $\frac{\tau_1 - \tau_2}{\tau_1}$, can be reached. But the engine may take in part of its supply of heat at temperatures below τ_1 and still act reversibly in the conversion of the heat so received into work, in which case the efficiency of the whole action will be less than $\frac{\tau_1 - \tau_2}{\tau_1}$ though the general formula $\frac{\tau - \tau_2}{\tau}$ is still applicable in respect of every separate portion of the heat, when proper values are assigned to τ.

The ideal steam-engine which we are now considering is a case in point. It takes in the greater part of its heat at τ_1, but some is taken in at temperatures ranging between τ_2 and τ_1. So far as actions occurring within the engine are concerned it is reversible. The amount of heat it converts into work is therefore to be found by calculating

$$\Sigma \frac{\delta Q (\tau - \tau_2)}{\tau},$$

where δQ represents any part of the heat taken in and τ the temperature at which it is taken in. The whole heat taken in, per lb. of working substance, is, first, the amount of heat which is required to heat the water from τ_2 to τ_1, namely, $h_1 - h_2$, which is taken in while the temperature is varying, and, second, the latent heat L, which is taken in at the temperature τ_1. Hence the whole amount of work done per lb. of working steam (expressed in thermal units) is

$$W = \int_{h_2}^{h_1} \frac{dh\,(\tau - \tau_2)}{\tau} + \frac{L_1(\tau_1 - \tau_2)}{\tau_1},$$

which may be written

$$W = \int_{h_2}^{h_1} dh - \tau_2 \int_{h_2}^{h_1} \frac{dh}{\tau} + \frac{L_1(\tau_1 - \tau_2)}{\tau_1}.$$

This gives $$W = h_1 - h_2 - \tau_2 \log_e \frac{\tau_1}{\tau_2} + \frac{L_1(\tau_1 - \tau_2)}{\tau_1} \quad\text{............(7)},$$

since dh may be taken as almost equal to $d\tau$, the specific heat of water being very nearly constant and equal to unity. We might for the same reason write $h_1 - h_2 = \tau_1 - \tau_2$ and express the result thus

$$W = (\tau_1 - \tau_2)\left(1 + \frac{L_1}{\tau_1}\right) - \tau_2 \log_e \frac{\tau_1}{\tau_2} \quad\text{..............(8)}.$$

This is the greatest amount of work which can be done, per lb. of steam, under ideally favourable conditions by an engine which takes steam from a boiler at temperature τ_1 and restores condensed water to the boiler at temperature τ_2. The result is interesting as affording a standard with which the performance of actual steam-engines may be compared (see Chapter V.)[1].

It is convenient to have a name for the cycle of operations here considered. Following a practice which has now become common we may call it the Clausius cycle.

The efficiency of a steam-engine working in this ideally favourable manner, but without compression, is to be found by dividing the above expression for W by the heat taken in per lb. of steam, which is

$$L_1 + h_1 - h_2.$$

As a numerical example, take the case of an ideal engine

[1] In finding numerical values of W from this equation the quantity $\frac{L}{\tau}$ may be conveniently taken from the table in the Appendix. It is the difference between the numbers given there under the headings ϕ_s and ϕ_w. (See § 87, below.)

working in this way, receiving steam at an absolute pressure of 160 lbs. per sq. inch, and condensing it at 60° F., with complete adiabatic expansion from the top to the bottom of this range. Here τ_1 is 824, τ_2 is 521 and L_1 is 858, in round numbers, and hence the expression for W gives 379 thermal units as the equivalent of the work done per lb. of steam. The supply of heat per lb. is 1165 thermal units. The efficiency is therefore 0·325. Compare this with the number 0·368 which represents the value of $\frac{\tau_1 - \tau_2}{\tau_1}$; namely, the efficiency of a reversible cycle completed by adiabatic compression as in the engine of § 67. The absence of adiabatic compression has in this case reduced the efficiency by nearly 12 per cent. This comparison shows what is lost by the partial misapplication of heat which results from letting the feed-water come into the boiler cold, to be heated by contact with the hot water already there, so that the portion of heat represented by $h_1 - h_2$ is taken in at temperatures lower than the top of the range.

The expression for W in Eq. 8 serves to show how much work it is ideally possible to get from 1 lb. of steam when the temperature of the boiler and the temperature of the condenser are assigned. But it is further useful as a standard with which we may compare the action of the steam cylinder, taken by itself, without reference to the boiler or to the condenser. For that purpose τ_1 may be understood as the temperature of the steam on reaching the cylinder and τ_2 the temperature of the steam on leaving the cylinder, although these temperatures may differ from those of the boiler and condenser respectively. Then the formula serves to show a limiting amount of work which the actual performance of the steam must be expected to fall short of, and furnishes a useful check on the results of engine trials.

80. Extension to the case of steam not initially dry. The result arrived at in the last paragraph may be readily extended to cases where the steam is not dry when the adiabatic expansion begins. Let q_1 be the dryness at this stage: then the heat taken in during evaporation is $q_1 L_1$ per lb. of working substance, but the heat taken in during the heating of the water up to τ_1 remains what it was before. The expression for the work done per lb. of feed-water (assuming complete adiabatic expansion

as before) is therefore found by substituting q_1L_1 for L_1 in equation (7) or (8), giving,

$$W = h_1 - h_2 - \tau_2 \log_e \frac{\tau_1}{\tau_2} + \frac{q_1L_1(\tau_1 - \tau_2)}{\tau_1} \quad\text{............(9).}$$

81. Derivation of the adiabatic equation from this result. This result may be applied to prove the equation for the adiabatic expansion of steam which was stated, without proof, in § 65. The whole heat taken in, per lb., in raising the water from any temperature τ_2 to τ_1 and in evaporating the fraction q_1 of it at the temperature τ_1 is

$$h_1 - h_2 + q_1L_1.$$

By expanding this mixture adiabatically to the temperature τ_2 and then condensing it, we get an amount of work equal (by the equation which has just been given) to

$$h_1 - h_2 + q_1L_1 - \frac{q_1L_1\tau_2}{\tau_1} - \tau_2 \log_e \frac{\tau_1}{\tau_2}.$$

Hence, subtracting the work done from the heat supplied we find that the heat rejected is

$$\frac{q_1L_1\tau_2}{\tau_1} + \tau_2 \log_e \frac{\tau_1}{\tau_2}.$$

But the only rejection of heat in the cycle takes place during the condensation at τ_2 after adiabatic expansion, and the amount of heat so rejected is

$$q_2L_2,$$

where q_2 is the dryness after adiabatic expansion to the temperature τ_2.

Hence
$$q_2L_2 = \frac{q_1L_1\tau_2}{\tau_1} + \tau_2 \log_e \frac{\tau_1}{\tau_2}$$

or
$$\frac{q_2L_2}{\tau_2} = \frac{q_1L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau_2}.$$

Now τ_2 may be any temperature lower than τ_1, for the adiabatic expansion might be stopped at any point along the curve and the cycle completed by condensing the mixture at the temperature it had then reached. Hence this equation serves to show in a perfectly general way the change of dryness which takes place during adiabatic expansion, and, dropping the second suffix, we may write it

$$\frac{qL}{\tau} = \frac{q_1L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau} \quad\text{....................(10),}$$

which is the same as equation (10) in § 65. It is to be noticed that in deriving this expression the specific heat of water has been treated as constant. The result is therefore (to a very small extent) inexact, especially at high temperatures.

82. **Entropy.** When a substance takes in or rejects heat it is said to change its *entropy*, the change of entropy being defined by the expression

$$\Sigma \frac{\delta Q}{\tau},$$

each element δQ of the heat taken in or rejected being divided by the absolute temperature which the substance had at the time. In dealing with entropy, just as in dealing with total heat, it is convenient to choose some arbitrary starting-point and reckon the entropy from that point as a zero. Thus in reckoning the entropy of steam at any temperature we may take the condition of water at 32° Fah. as a convenient datum and calculate $\Sigma \frac{\delta Q}{\tau}$ from that, calling the value so calculated the entropy of the steam. Entropy will be denoted by ϕ: in giving it numerical values it is to be reckoned per unit mass of the substance[1].

It follows from this definition that when any substance is going through an adiabatic process its entropy does not change. Further we have seen (§ 78) that when a substance is carried through a complete reversible cycle

$$\Sigma \frac{\delta Q}{\tau} = 0$$

when the whole cyclic operation is considered. Hence when the cycle is complete the entropy of the substance, as well as its pressure, temperature, volume and internal energy, has returned to the value which it had at the beginning of the cycle.

Consider now a cycle consisting of two isothermal and two adiabatic operations, fig. 20. In passing from a to b by the isothermal line τ_1 the substance gains entropy $\frac{Q_1}{\tau_1}$, where Q_1 is the heat taken in during this operation. Along the adiabatic line from b to c there is no change of entropy. In the isothermal

[1] The name Entropy was first used by Clausius. Rankine calls ϕ the "Thermodynamic Function."

line *cd* the entropy is reduced by $\frac{Q_2}{\tau_2}$, and from *d* to *a* there is again no change of entropy. Now $\frac{Q_1}{\tau_1} = \frac{Q_2}{\tau_2}$, which means that the entropy

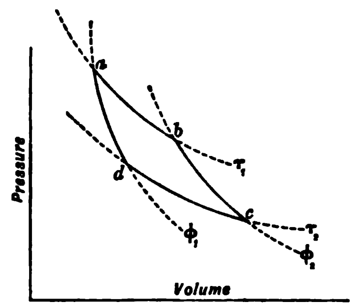


FIG. 20.

changes by the same amount whether we pass from one adiabatic line *ad* to another adiabatic line *bc* by one isothermal path *ab* or by any other isothermal path *dc*. And moreover the change of entropy between one adiabatic and another will be the same whether the cross-path be isothermal or not, for a curve expressing any relation between *P* and *V* may be regarded as made up of a succession of minute isothermal and adiabatic elements, and the change of entropy along such a curve is the sum of the changes which occur during the isothermal elements of the process, and is still equal to $\frac{Q}{\tau}$ for any single isothermal path between the same pair of adiabatic lines.

We see, then, that not only is there no change of entropy during an adiabatic process, but there is a perfectly definite change of entropy when a given substance passes from one adiabatic line to another, by whatever path. Just as isothermal lines are lines of uniform temperature so adiabatic lines are lines of uniform entropy, and just as isothermal lines can be distinguished by numbers τ_1, τ_2, etc. denoting the particular temperature for which each is drawn, so adiabatic lines can be distinguished by numbers ϕ_1, ϕ_2, etc. denoting the particular value of the entropy on each. From this point of view adiabatic lines are often called isentropic lines. The conception of entropy as that characteristic of a substance which does not change during adiabatic expansion or compression is of considerable service in problems relating to

heat-engines. We proceed to show some of the uses to which this notion may be put.

83. Entropy of Steam: Derivation of the Adiabatic Equation. Reckoning from water at any initial temperature τ_0 the entropy of steam (taken wet, for greater generality)

$$\phi = \int_{\tau_0}^{\tau_1} \frac{dh}{\tau} + \frac{q_1 L_1}{\tau_1}.$$

The first term represents the entropy which is acquired during the heating of the water from τ_0 to τ_1, which is the temperature of evaporation, and the second term represents what is acquired during evaporation, q_1 being the dryness of the steam. Treating the specific heat of water as unity we can write $d\tau$ for dh; then integrating,

$$\phi = \log_e \tau_1 - \log_e \tau_0 + \frac{q_1 L_1}{\tau_1} \quad \text{.................} \quad (11).$$

Now in adiabatic expansion we have

$$\phi = \text{constant},$$

and hence if the steam be expanded adiabatically to any temperature τ

$$\log_e \tau - \log_e \tau_0 + \frac{qL}{\tau} = \log_e \tau_1 - \log_e \tau_0 + \frac{q_1 L_1}{\tau_1},$$

from which

$$\frac{qL}{\tau} = \frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau_2},$$

which is the adiabatic equation of § 65, already derived by another and longer method in § 81.

84. Entropy-Temperature Diagrams. The familiar way to represent graphically those changes which a working substance undergoes in the action of a heat-engine is to draw the indicator diagram, which shows pressure in relation to volume. Another way is to draw a diagram showing the relation of the temperature of the substance to its entropy. Diagrams of this kind form an interesting and often useful alternative to the ordinary indicator diagram[1]. Let $\delta\phi$ be the small change in entropy which a

[1] Entropy-Temperature diagrams were described along with other graphic methods in thermodynamics by Professor J. Willard Gibbs (*Trans. of the Connecticut Acad. of Sciences*, Vol. II. 1873, p. 309). Their application to steam-engine problems is mainly due to Mr J. Macfarlane Gray (see *Proc. Inst. Mech. Eng.* 1889, p. 399). Professor Boulvin (*Cours de Mécanique appliquée : Théorie des Machines thermiques*) ascribes their earliest use to M. Th. Belpaire (*Bulletin de l'Académie royale de Belgique* 1872, v. 34).

substance undergoes when it takes in any small quantity δQ of heat at any temperature τ. By the definition of entropy

$$\delta\phi = \frac{\delta Q}{\tau},$$

whence $$\tau\delta\phi = \delta Q$$

and $$\int \tau d\phi = \int dQ \quad \text{.......................... (12)},$$

the integration being performed between any assigned limits. Now if a curve be drawn with τ and ϕ for ordinates, $\int\tau d\phi$ is the area under the curve. This by the above equation is equal to $\int dQ$, in other words, the area under any portion of the entropy-temperature curve is equal to the whole quantity of heat taken in while the substance passes through the states which that portion of the curve represents. Let ab, fig. 21, be any portion of the curve of ϕ and τ. The area of the cross-hatched strip whose breadth is $\delta\phi$ and height τ, is $\tau\delta\phi$, which is equal to δQ, the heat taken in during the small change $\delta\phi$. The whole area *mabn* or $\int\tau d\phi$ between the limits a and b is the whole heat taken in while the substance changes from the state represented by a to the state represented by b. Similarly, in changing from state b to state a by the line ba the substance rejects an amount of heat which is measured by the area *bamn*. The base line *ox* corresponds to the absolute zero of temperature.

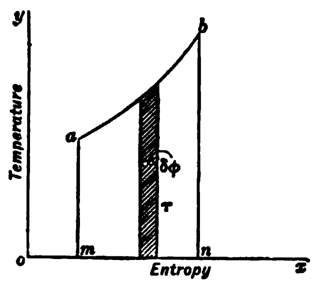


Fig. 21. Entropy-Temperature Curve.

When an entropy-temperature curve is drawn for a complete cycle of changes it forms a closed figure, since the substance returns to its initial state. To find the area of the figure we have to integrate throughout the complete cycle, when

$$\int\tau d\phi = Q_1 - Q_2,$$

Q_1 being the heat taken in and Q_2 the heat rejected. But the difference between these is the heat converted into work, hence

$$\int\tau d\phi = W \quad \text{.......................... (13)},$$

when the integration extends round a complete cycle and W is expressed in thermal units. Thus entropy-temperature diagrams have the important property in common with pressure-volume

diagrams that the enclosed area measures the work done in a complete cycle.

Isothermal lines on an entropy-temperature diagram are straight lines parallel to ox whatever be the working substance: adiabatic lines are straight lines parallel to oy, being lines of constant entropy. Hence Carnot's cycle, whether with air or steam or any other substance, would be represented by a rectangle $abcd$, fig. 22, in which the heat received

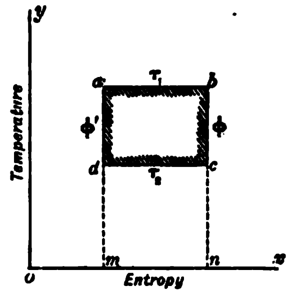


FIG. 22. Carnot's Cycle on the Entropy-Temperature Diagram.

$$Q_1 = \text{area } abnm = \tau_1 (\phi - \phi'),$$

heat rejected

$$Q_2 = \text{area } cdmn = \tau_2 (\phi - \phi'),$$

and work done

$$W = \text{area } abcd = (\tau_1 - \tau_2)(\phi - \phi'),$$

ϕ being the entropy in the adiabatic process of expansion and ϕ' the entropy in the adiabatic process of compression. The efficiency is

$$\frac{\text{area } abcd}{\text{area } abnm} = \frac{\tau_1 - \tau_2}{\tau_1}.$$

85. Entropy-Temperature Diagram for Steam: application to ideal steam-engine working without compression but with complete expansion. A more interesting example of the use of entropy-temperature diagrams is given by the engine of § 71 using the Clausius cycle of operations. In that cycle, after complete adiabatic expansion from τ_1 to τ_2 the steam is condensed isothermally at τ_2, and is then returned as water to the boiler. In drawing the diagram for the cycle we shall begin at the point where the water, at τ_2, is about to be heated. Reckoning from some standard (lower) temperature τ_0 and dealing throughout with 1 lb. of the working fluid, we have

$$\text{Entropy of water at any temperature } \tau = \int_{\tau_0}^{\tau} \frac{dh}{\tau} = \int_{\tau_0}^{\tau} \frac{\sigma d\tau}{\tau},$$

where σ is the specific heat of water. The specific heat is equal to unity at low temperatures and becomes only a very little more than unity at high temperatures. Neglecting this small change, we may write

$$\text{Entropy of water} = \int_{\tau_0}^{\tau} \frac{d\tau}{\tau} = \log_e \tau - \log_e \tau_0.$$

which relates to any stage in the heating of the feed-water from τ_2 to τ_1. The first part of the diagram is therefore a logarithmic curve, *ab*, fig. 23, where $\tau_a = \tau_2$, $\tau_b = \tau_1$, $\phi_a = \log_e \tau_2 - \log_e \tau_0$, and $\phi_b = \log_e \tau_1 - \log_e \tau_0$. Hence $\phi_b - \phi_a$ or $mn = \log_e \tau_1 - \log_e \tau_2$. It is a matter of indifference, in the drawing of the diagram, at what distance the origin is taken to the left of m; in other words, what

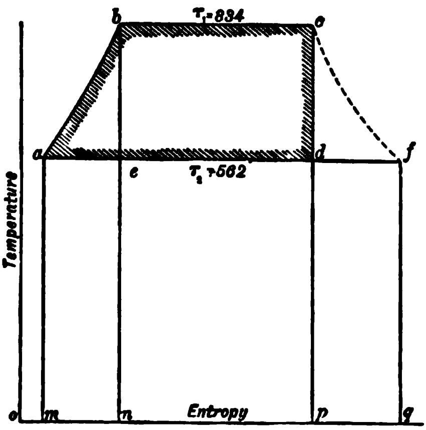


FIG. 23. Entropy-Temperature Diagram for Steam.

value of τ_0 is taken as a datum in reckoning the entropy. In the example sketched in the figure the entropy of water at 32° Fah. is reckoned to be zero: τ_a is taken to be 562 and τ_b to be 834; τ_a therefore corresponds to a steam pressure of 1 lb. per square inch and τ_b to a pressure of 180 lbs. per sq. inch. At *b* steam begins to be formed, and *bc* is the change of entropy which the substance undergoes in passing from water to steam at the constant temperature τ_1. *bc* is therefore equal to $\frac{L_1}{\tau_1}$, assuming the evaporation to be complete. If the evaporation were incomplete, *bc* would be equal to $\frac{q_1 L_1}{\tau_1}$. The adiabatic process of complete expansion down to the temperature τ_2 is represented by *cd*, and *da* is the process of condensation which completes the cycle.

The heat taken in during the warming of the feed-water is the area *mabn*. The heat taken in during evaporation is *nbcp*. The work done is the enclosed area *abcda*. The heat rejected is *pdam*. Of the heat taken in during the process of evaporation, namely, the area *bp*, the part measured by the area *bd* is converted

into work: it represents the fraction $\frac{be}{bn}$ or $\frac{\tau_1 - \tau_2}{\tau_1}$ of the whole, as we should expect. Of the heat taken in during the process of warming the feed-water a smaller fraction is converted into work, namely, the fraction $\frac{abe}{mabn}$. This is because the heat is less advantageously supplied during this operation, the temperature being then less than τ_1. An engine going through Carnot's cycle would have the diagram *ebcd*. The present engine does more work (by the area *abe*), but to do this it has to take in a more than proportionally larger amount of heat and is therefore less efficient. It will be seen that the diagram exhibits in a very simple way results which we have already arrived at by other routes.

Further, let a curve *cf* be drawn such that the distance from any point in *ab* to it, measured horizontally (that is, parallel to *op*), is equal to the value of $\frac{L}{\tau}$ corresponding to that point. Thus let *af* be equal to $\frac{L_2}{\tau_2}$, *f* being the point where *ad* produced meets this curve. Then the dryness of the steam after the process of adiabatic expansion represented by *cd* is given by the fraction $\frac{ad}{af}$. This follows from the fact that if the steam were perfectly dry at τ_2 the heat given out during its condensation would be equal to the area *qfam*, whereas the heat actually given out is equal to *pdam*. In other words, the former area is L_2 and the latter is q_2L_2, q_2 being the dryness when the stage *d* is reached, whence $q_2 = \frac{ad}{af}$. In the same way a straight line drawn horizontally from any point *i* in *cd* (fig. 24) to meet the curves *cf* and *ab* is

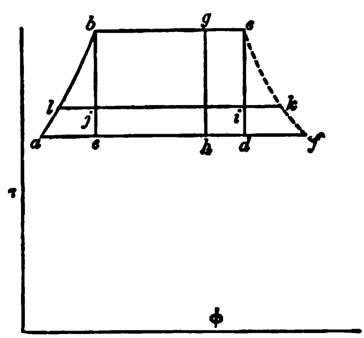


FIG. 24.

divided by cd into segments il and ik. These are proportional to the quantities of steam and water respectively which make up the working substance when the expansion has advanced as far as the point i. In other words, the dryness $q = \frac{li}{lk}$. The temperature-entropy diagram thus affords a convenient method of finding q graphically at any stage in adiabatic expansion.

Further, suppose the steam has not been dry when adiabatic expansion begins. This state of things is represented on the diagram by making the horizontal line from b terminate at a point g such that $bg = \frac{q_1 L_1}{\tau_1}$: in other words, $\frac{bg}{bc} = q_1$. The line gh now represents the process of adiabatic expansion and the construction just described is still applicable to find q at any stage. Thus at h, $q = \frac{ah}{af}$ and $\frac{hf}{af}$ is the proportion then present as water.

Again, reverting to the Carnot cycle of fig. 14, § 67, we can use the entropy-temperature diagram to determine the point at which condensation at τ_2 must be stopped in that cycle in order that adiabatic compression may bring the substance to the state of water at τ_1. The process of compression required for this is eb (fig. 23 or 24), and hence compression must begin when the proportion of steam still uncondensed is $\frac{ae}{af}$. Similarly the fraction $\frac{lj}{lk}$ measures the dryness at any stage j of this adiabatic compression.

86. Application of the entropy-temperature diagram to the case of superheated steam. The entropy of steam superheated to any temperature τ' is to be found by adding to the expression for the entropy in the saturated state the term

$$\int_{\tau_1}^{\tau'} \frac{\kappa d\tau}{\tau},$$

where κ is the specific heat of the steam during superheating, that is to say, the amount of heat required to raise 1 lb. of the steam 1° Fah. when its temperature exceeds τ_1 the temperature of saturation. In the absence of more definite knowledge of what happens during superheating κ is generally assumed to be roughly constant. Its value is probably not far from 0·5 when there is much

superheating[1]; and when the process of superheating is performed at constant pressure,—a condition which applies, for instance, when steam is superheated by passing through a coil of pipe in a hot flue or furnace on its way from the boiler to the engine. The addition to the entropy may then be written, approximately,

$$\kappa\int_{\tau_1}^{\tau'}\frac{d\tau}{\tau}=\kappa(\log_e\tau'-\log_e\tau_1).$$

This allows the entropy-temperature line to be extended as in fig. 25,

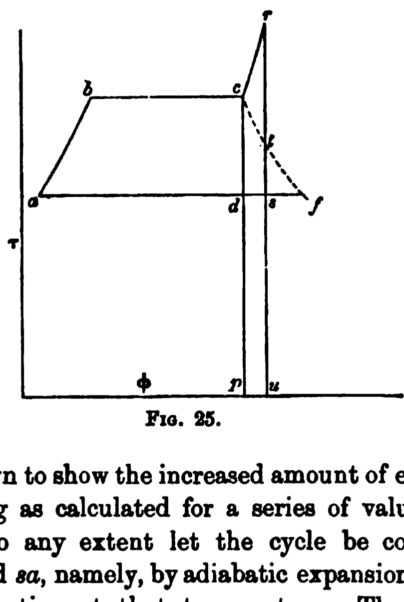


FIG. 25.

where cr is drawn to show the increased amount of entropy produced by superheating as calculated for a series of values of τ'. After superheating to any extent let the cycle be completed by the processes rs and sa, namely, by adiabatic expansion to temperature τ_2 and condensation at that temperature. The diagram shows that, in consequence of superheating, the work done by the substance is increased by the area $dcrs$, while the heat taken in is increased by $pcru$. The efficiency is slightly increased, since this additional heat is received at temperatures somewhat higher than those at which the other portions of the heat were received. But unless superheating be carried very far the extra supply of heat is too small a part of the whole to make any large difference in the efficiency of the ideal engine we are dealing with here. In the case sketched in

[1] When the amount of superheating is small κ may be expected to be greater than this, especially when the pressure is high. It is not strictly constant, the amount of heat taken in per degree being greater at the beginning of the superheating than when the temperature is considerably raised. (See § 90.)

fig. 25 the steam is supposed to be superheated as much as 200° above the boiler temperature, but the diagram shows that even this makes but little improvement in the ideal efficiency. In real engines superheating does make a marked difference, but its influence is indirect, and proceeds from the fact that it tends to prevent the steam from being condensed by contact with the metal of the cylinder and piston. This effect of superheating will be considered in the next chapter. Nothing of the kind takes place in the ideal case now dealt with, because here we postulate adiabatic expansion, or, in other words, a perfectly non-conducting cylinder and piston.

It would evidently be fallacious to suppose that when superheating is applied to the steam of the ideal engine the increased range of temperature implies anything like a corresponding gain of efficiency, for the chief part of the heat is still taken in at the temperature of saturation, and its value for conversion into work depends on the temperature at which it is taken in, not upon the temperature to which the working substance is subsequently raised.

In the diagram, fig. 25, the adiabatic line *rs* shows by its intersection of the curve *cf* at *t* the stage in the expansion at which the steam will cease to be superheated. At this point *t* it is dry and saturated: as the expansion proceeds it becomes wet, and at the end of expansion the condensed part is $\frac{sf}{af}$ of the whole. The extent to which superheating has to be carried if the steam is just to be dry, and no more, at the end of expansion, is readily found by drawing a vertical line through *f* to meet the continuation of the curve *cr*.

87. Values of the Entropy of Water and Steam. In applying this useful graphic method to the investigation of particular cases in the expansion of steam it is convenient to have an entropy-temperature chart for water and steam drawn on section-paper throughout the range of pressures which are found in practice: the construction for particular cases is then readily made by adding horizontal straight lines to correspond with the formation and condensation of the steam, while any adiabatic process is represented by a vertical line.

Such a diagram, carefully drawn to scale, is shown in fig. 26.

The curves extend throughout the whole range of Table I. and more than cover the useful range of pressure. The curve on the left marked "water" shows the relation of entropy to temperature before steam begins to form: the curve on the right marked "steam" shows the same relation when all the water is converted into steam. The horizontal distance between the two curves at any point, or $\phi_s - \phi_w$, represents the gain of entropy which occurs while the water is changing into steam $\left(\text{namely } \frac{L}{\tau}\right)$. An extension of the diagram to the left might be drawn, in the form of a horizontal line at 32° Fah. to show the change of entropy when

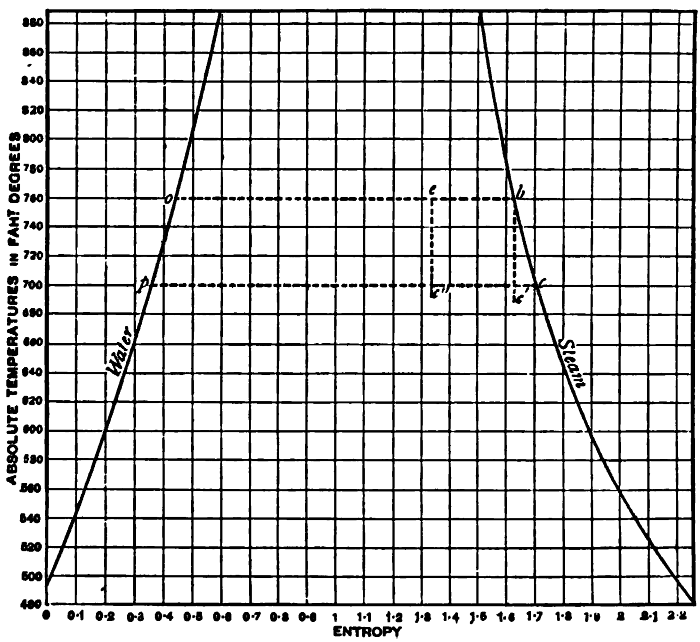


FIG. 26. Entropy of Water and Steam.

ice melts; but this would have no application to our present purpose. The numerical values of the entropy relate to 1 lb. of water or steam and are reckoned from water at 32° Fah. In calculating the entropy of water allowance has been made for the increased specific heat of water at high temperatures. The diagram has been drawn by calculating the entropy of water (ϕ_w)

and of steam (ϕ_s) for certain of the points in Table I., from the data furnished by Regnault's experiments; the values so calculated are given in Table II. below. A more extended table showing the entropy of water and steam at various pressures will be found, along with other properties of steam, in the Appendix.

TABLE II. *Entropy of Water and Steam.*

Temperature.		Pressure, lbs. per sq. in.	$\frac{L}{\tau}$	Entropy.	
$t°$ Fah.	τ.			Water. ϕ_w	Steam. ϕ_s
32	493	0·085	2·215	0	2·215
95	556	0·806	1·885	0·121	2·006
149	610	3·62	1·656	0·214	1·870
212	673	14·70	1·436	0·312	1·748
257	718	33·71	1·301	0·378	1·679
302	763	69·21	1·182	0·440	1·622
347	808	129·8	1·076	0·499	1·575
392	853	225·9	0·981	0·554	1·535
428	889	336·3	0·912	0·598	1·510

From this diagram and from the knowledge of the relation of pressure to volume in saturated steam which is furnished by the table in the Appendix, it is easy to determine what proportion of water will be present at any stage in adiabatic expansion or compression, and hence to draw the ordinary indicator diagram or pressure-volume curve for an adiabatic process. Thus let *BCD*, fig. 27, be a portion of the pressure-volume curve for *saturated* steam. To draw the adiabatic curve from any assigned point *B*, refer to the table to find the temperature which corresponds to the assigned pressure at *B*, and draw a horizontal line *ob* at that temperature in the entropy diagram (fig. 26). If the steam is assumed to be dry at *B*, draw a vertical line *bc′* through *b*. Taking any lower pressure draw the horizontal line *NC* in the pressure-volume diagram (fig. 27), refer to the table for the corresponding temperature, and then draw the line *pc* for that temperature in fig. 26. Measure the ratio $\frac{pc'}{pc}$. This is the dryness at *C*. Take a point *C′* in *NC* (fig. 27) such that $\frac{NC'}{NC}=\frac{pc'}{pc}$. Then *C′* is (sensibly) a point on the adiabatic curve. The same construction is

to be repeated to find as many points as are sufficient to let the curve be drawn. If the steam is wet to begin with, the initial volume of 1 lb. will have some value ME less than MB and the curve of

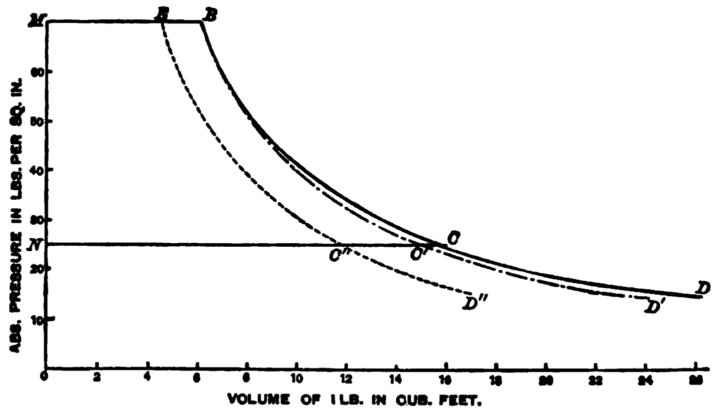


FIG. 27. Pressure-volume curves in adiabatic expansion.

adiabatic expression starts from E. It is found in that case by taking e in fig. 26 so that $\frac{oe}{ob} = \frac{ME}{MB}$ (the initial dryness), drawing the vertical ec'', and taking C'' in fig. 27 so that $\frac{NC''}{NC} = \frac{pc''}{pc}$, this ratio being the dryness after adiabatic expansion has brought the pressure of the mixture down to the level of pressure NC; C'' is then a point in the required curve. The curve $EC''D''$ has been sketched in this way to show the adiabatic expansion of steam containing 25 per cent. of moisture to begin with.

In the example sketched the pressure at M is 70 lbs. per square inch, and at N it is 25 lbs. per square inch. The lines ob and pc are drawn in fig. 26 at the corresponding levels of temperature.

Not the least merit of the entropy-temperature diagram as a means of representing graphically the cycle of operations in a heat-engine is that it shows the heat taken in and the heat rejected, as well as the work done, and so allows estimates of efficiency to be made by inspection of the diagram itself. The advantage, for instance, which results from raising the initial pressure of the steam is readily shown in a diagram such as fig. 23 by drawing horizontal lines at temperatures corresponding to the

8—2

initial pressures which are to be compared, and vertical lines through the points where they meet the entropy curve of saturated steam (*cf*), the vertical lines being continued to meet the base, which is the absolute zero of temperature. Comparison of the enclosed areas then shows that while the heat taken in is but slightly increased with higher boiler pressure there is a more considerable gain of work, a result which is of course to be expected from the fact that the general temperature of reception of the heat is raised.

If a vertical line such as ec'' in fig. 26 be drawn to represent the adiabatic expansion of a mixture of steam and water, it is clear from the diagram that when e is chosen at less than a certain distance from o, that is to say, when there is a certain degree of initial wetness, the mixture will become drier as it begins to expand, instead of wetter as is the case when the initial proportion of water is less. In the region of ordinary working pressures the "water" and "steam" curves of fig. 26 have nearly equal inclinations to the vertical line which represents an adiabatic process. Hence if such a line be drawn starting from a point midway between the two curves it will continue to lie nearly midway between them: in other words, if there is about 50 per cent. of water present at the beginning of adiabatic expansion, nearly the same percentage will be found as the expansion goes on. When the steam is much wetter than this to begin with, adiabatic expansion makes it drier.

It will be shown later that the entropy-temperature diagram is also of service in exhibiting the changes of dryness which occur in real steam-engines, where the action is by no means adiabatic.

88. Entropy-temperature diagram for Steam used Non-expansively. By way of contrast with the cases treated in §§ 85 and 86, we may draw the entropy-temperature diagram for

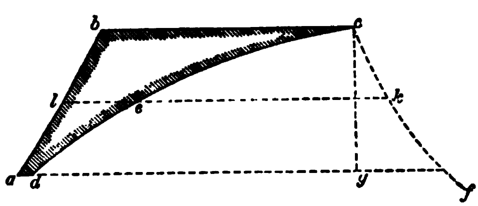


FIG. 28. Entropy-temperature diagram of Steam used Non-expansively.

a steam-engine working without expansion. The four steps of the cycle have been stated in § 69, and the volume-pressure diagram is drawn there (fig. 15). In the entropy-temperature diagram (fig. 28) we have the four corresponding lines *ab*, *bc*, *cd*, *da*. *ab* is the heating of the water from τ_2 to τ_1. *bc* is the conversion of the water into steam, *cd* the partial condensation which takes place when the cold body is applied, the piston meanwhile remaining at the end of its forward stroke, and *da* is the remainder of the condensation, which occurs while the piston is pressed in, the cold body being still applied. *cd* is a line of *constant volume*, for throughout the change which it represents the substance remains in the cylinder and there is no movement of the piston. To find points in *cd*, draw the saturation curve *cf* as in former examples and at any temperature τ intermediate between τ_1 and τ_2 draw the line *lk*. We have to divide *lk* in a point *e* such that $\frac{le}{lk}$ shall represent q, the dryness of the steam at the time its temperature has fallen to τ. The dryness q is determined by the consideration that qV is sensibly equal to V_1, where V is the volume of 1 lb. of saturated steam at τ and V_1 is the volume originally occupied by 1 lb. before the process of condensation began. Throughout the operation *cd* the volume of the substance remains unchanged and equal to V_1. Hence $q=\frac{V_1}{V}$, and *e* is found by making

$$le=\frac{V_1}{V}\,lk.$$

The work which is lost through the absence of adiabatic expansion is the area *cgd*. In the example sketched the initial pressure is 180 lbs. per sq. inch, and the pressure during the return stroke *da* or the "back pressure" is 3 lbs. per square inch. In other words, τ_1 is taken as 834 and τ_2 as 603.

89. Incomplete expansion. The case of incomplete expansion admits of similar treatment. Let adiabatic expansion be carried on until, at the end of the stroke, the temperature has fallen to the level indicated by *c'* in the entropy-temperature diagram, fig. 29. This process is represented by the line *cc'*. Then let the steam be suddenly cooled by applying the cold body. The constant-volume curve *c'd* shows this cooling; after which the

return stroke takes place, which is shown by *da*. To draw the curve *c′d* take *e* at any level, such that

$$le = \frac{q'V'}{V} lk,$$

where V' is the volume of 1 lb. of saturated steam at the temperature corresponding to c', and q' is the dryness at c', which is equal to $\frac{mc'}{mn}$.

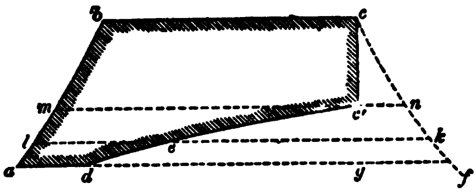


FIG. 29. Entropy-temperature diagram showing incomplete expansion of Steam.

In the example sketched in fig. 29 the pressure is reduced by adiabatic expansion in the operation *cc′* from 180 to 20 lbs. per square inch, and the back pressure is 3 lbs. per square inch as in fig. 28.

In dealing with the process of sudden condensation represented by the line *cd* in fig. 28 and *c′d* in fig. 29 we have supposed, to simplify the statement, that the steam is retained in the cylinder and the cold body is applied to it. But it makes no difference if the steam be allowed to escape into a separate vessel, to be condensed there. Just the same amount of work is done, for the pressure on the piston is the same in that case as in the other. Hence the area of the entropy-temperature diagram is unaffected, and since that is true whatever be the value of τ_2 the form of the curve *cd* or *c′d* is unchanged[1].

90. The Total Heat of Superheated Steam. Reference was made in § 63 to Rankine's method of calculating the total

[1] The constant-volume curve *cd* or *c′d* in the entropy-temperature diagram may be more conveniently drawn as follows by an application of equation (8) of § 76. Let U represent the volume of the mixture of steam and water at any stage in the process of condensation, the temperature then being τ. Let λ represent the heat which would be given out if the condensation of the mixture were completed at the temperature τ. Then by that equation

$$U - \omega = \frac{J\lambda}{\tau} \frac{d\tau}{dP},$$

heat of very highly superheated steam, or "steam-gas," by the formula

$$H' = 1092 + 0{\cdot}48\,(t' - 32).$$

This formula is arrived at as follows. It is taken as established by the experiments of Regnault that if the pressure of a vapour is low it behaves like a perfect gas from the first, when it is being superheated. Thus water vapour at 32° F. when superheated under constant pressure is taken as having a constant specific heat, of the value 0·48. Hence the total heat of superheated steam when made by evaporating water at 32° F. and then superheating it at constant pressure to any temperature t' will be that given by the above formula, for 1092 is simply the total heat of the vapour at 32° F. Now suppose t' is very high. The steam will then continue to behave like a perfect gas when its pressure is changed. Suppose the pressure to be increased isothermally from the first value to any other value; we have to prove that the total heat is not changed by this process. In other words that, provided the temperature is high enough to make steam act like a perfect gas, the total heat is independent of the pressure. If this be so it is clear that the expression for H' applicable to steam-gas at the low pressure which corresponds to saturation at 32° will also be applicable to steam-gas at any other pressure, when t' is the same in both cases. To prove that the total heat of steam-gas is independent of the pressure we consider an imaginary cycle in which the pressure is varied from any value p_1 to any other value

ω being the volume when the substance is all water. Hence

$$\frac{\lambda}{\tau} = \frac{U - \omega}{J}\frac{dP}{d\tau}.$$

But $\frac{\lambda}{\tau}$ is the length le, if the line le be drawn at the level τ, and U is the volume of the cylinder, which is constant. We therefore have

$$le \propto \frac{dP}{d\tau},$$

a relation which allows le at any level of temperature to be readily determined when the values of $\frac{dP}{d\tau}$ for saturated steam are known. These may be found by measurement of the slope of the pressure-temperature curve, or approximately from the table of P and τ in the Appendix by dividing small differences of pressure by corresponding differences of temperature.

The method described in the text of drawing the constant-volume curve is given by Professor Cotterill in the second edition of his *Treatise on the Steam-Engine*, p. 302. The examples sketched here (figs. 28 and 29) are drawn to scale.

p_2 while the temperature t' does not change. First, let steam be formed at any pressure p_1 and superheated to the temperature t' (absolute τ'), which is assumed to be so high that the steam is then sensibly a perfect gas, however great be the pressure. Call H_1' the total heat taken in during this operation, and v the volume of the superheated steam. The work done in producing the steam in this condition is p_1v_1 (neglecting the small initial volume of the water). Then let the steam expand isothermally (still remaining very highly superheated) to some other volume v_2 and pressure p_2. During this expansion it acts, by assumption, like a perfect gas; hence $p_1v_1 = p_2v_2$ and heat is taken in equal to the work done, namely,

$$\int_{v_1}^{v_2} pdv \text{ or } c\tau_1' \log_e \frac{v_2}{v_1}.$$

Next suppose the steam to be condensed under the constant pressure p_2. It gives out a quantity of heat H_2' which is the total heat of superheated steam at temperature t' when formed under constant pressure p_2, and this is to be proved equal to H_1'. During this condensation work is spent on the steam, to the amount p_2v_2. The cycle is completed by raising the pressure of the condensed water from p_2 to p_1. Now taking the cycle as a whole, since

Heat taken in + work spent on the gas = heat given out
+ work done by the gas,

$$H_1' + c\tau' \log_e \frac{v_2}{v_1} + p_2v_2 = H_2' + p_1v_1 + c\tau' \log_e \frac{v_2}{v_1}.$$

But $$p_1v_1 = p_2v_2,$$

and hence $$H_1' = H_2'.$$

That is to say, the total heat of steam-gas is the same for the same temperature, under whatever pressure the steam has been formed, and therefore also whether the pressure during formation has been constant or not. Hence Rankine's formula ought to be applicable when the amount of superheating is very great, provided the experimental basis is correct that saturated steam at 32° F. is virtually a perfect gas, and provided the constants are right.

At such pressures, however, as are met with in steam-engine practice it would be necessary to carry superheating much further than it is actually carried in order to make the steam behave like

a perfect gas. Hence Rankine's formula is seldom, if ever, applicable as a means of finding the total heat. In ordinary cases the value given by it would be unduly great.

As an illustration, take the case of steam saturated at 212° F and superheated under constant pressure through 108° to 320° F. If the above formula were then applicable, H' would be 1230·2. In the saturated state H is 1146·6. Hence the heat taken in during superheating would, on this reckoning, be 83·6 thermal units, or on the average 0·77 units per degree, a quantity much greater than the commonly accepted constant 0·48. In such a case Rankine's formula no doubt largely over-estimates the total heat. If the superheating were carried further, say to 572° F., the average amount of heat required in the 360° of superheating would be 0·57 units per degree, if Rankine's H' were then a correct measure of the total heat. But his formula is not strictly applicable even then, and if we were dealing with steam of higher pressure it would be still less applicable. On the other hand, the usual assumption that the specific heat may be taken as constant in the superheating of high-pressure steam is probably erroneous, and it may be expected that experiment will show the first stage of superheating to require more heat per degree than the later stages[1].

91. Entropy diagram for engine working with steam saturated throughout expansion. This case has been mentioned in § 72 as the limiting case in a jacketed engine, when the jacket keeps the steam wholly dry during expansion. The entropy diagram is then the figure *abcf* (fig. 23), assuming the expansion to be complete. The saturation curve *cf* represents the process of expansion, and the area *pcfq* is the heat supplied by the jacket, H_J. The work done W is readily calculated from the area of the diagram, the width of which at any height is $\frac{L}{\tau}$. Hence

$$W = \int_{\tau_2}^{\tau_1} \frac{L}{\tau}\, d\tau.$$

[1] On this subject reference should be made to a paper by Professor Osborne Reynolds, "On methods of determining the dryness of steam and the condition of steam-gas," *Proc. of the Manchester Phil. Soc.* Nov. 1896. See also Ewing and Dunkerley, "On the specific heat of superheated steam," *Rep. Brit. Assoc.* 1897. It may be added that Regnault's experiments deal only with differences in the total heat of steam that is much and little superheated and do not show what is the specific heat for small amounts of superheating, even in the case of low-pressure steam.

To integrate this we may express L (as in § 59, Eq. 7) in the form

$$L = a - b\tau.$$

Then
$$W = \int_{\tau_2}^{\tau_1} \frac{a - b\tau}{\tau}\, d\tau = a \log_e \frac{\tau_1}{\tau_2} - b\,(\tau_1 - \tau_2).$$

When British thermal units are used a is 1437 and b is 0·7. This expression for W may be compared with the one given in § 72, and will be found to give the same values.

To find the heat supplied by the jacket, H_j, we have

$$\text{Total heat received} = L_1 + h_1 - h_2 + H_j,$$

$$\text{Heat rejected} = L_2.$$

Hence
$$H_j = W - (L_1 - L_2) - (h_1 - h_2)$$
$$= a \log_e \frac{\tau_1}{\tau_2} - (h_1 - h_2).$$

The efficiency is
$$\frac{a \log_e \frac{\tau_1}{\tau_2} - b\,(\tau_1 - \tau_2)}{a \log_e \frac{\tau_1}{\tau_2} + L_1}.$$

As a numerical example, to be compared with the example of adiabatic expansion given in § 79, suppose τ_1 to be 824 (pressure 160 lbs.) and τ_2 to be 521 (condensation at 60° F.). Then, in thermal units, $a \log_e \frac{\tau_1}{\tau_2}$ is 659, H_j is 353 and W is 447. The total heat taken in during the cycle is 1517 units, and the efficiency is 0·295, or about nine per cent. less than the efficiency in the corresponding cycle with adiabatic expansion. The reduced efficiency is of course due to the fact that the heat supplied by the jacket reaches the working steam when its temperature has fallen below the top of the range.

92. Entropy-temperature diagrams in engines using a Regenerator. An engine such as Stirling's, which substitutes the use of a regenerator for the adiabatic expansion and compression in Carnot's cycle, has an entropy diagram of the type shown in fig. 30. The isothermal operation of taking in heat at τ_1 is represented by ab; bc is the cooling of the substance from τ_1 to τ_2 in its passage through the regenerator, where it deposits heat: cd is the isothermal rejection of heat at τ_2; and da is the restoration of heat by the regenerator while the substance

passes through it in the opposite direction, by which the temperature is raised from τ_2 to τ_1. Assuming the action of the regenerator to be ideally perfect, *bc* and *ad* are precisely similar curves whatever be their form. The area of the figure is then equal to the area of the rectangle which would represent the ordinary Carnot cycle (fig. 22). The equal areas *pbcq* and *ndam* measure the heat stored and restored by the regenerator.

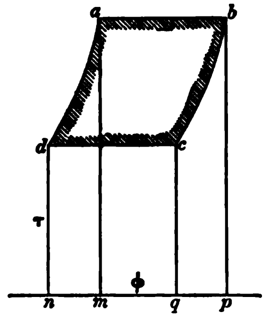


FIG. 30. Entropy-temperature diagram of perfect engine using a Regenerator.

When the working substance is air and the regenerative changes take place either under constant volume, as in Stirling's engine, or under constant pressure, as in Ericsson's, so that the specific heat K is constant, *ad* and *bc* are logarithmic curves with the equation

$$\phi = \int \frac{K d\tau}{\tau} = K \log_e \tau,$$

K being K_v in one case and K_p in the other.

93. Joule's Air-Engine. A type of air-engine was proposed by Joule which, for several reasons, possesses much interest.

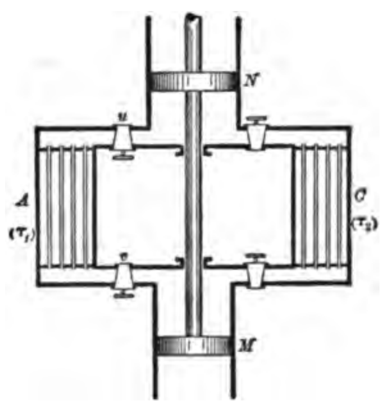


FIG. 31. Joule's proposed Air-Engine.

Imagine a chamber C (fig. 31) full of air (temperature τ_2), which is kept cold by circulating water or otherwise; another chamber

A heated by a furnace and full of hot air in a state of compression (temperature τ_1); a compressing cylinder M by which air may be pumped from C into A, and a working cylinder N in which air from A may be allowed to expand before passing back into the cold chamber C. We shall suppose the chambers A and C to be large, in comparison with the volume of air that passes in each stroke, so that the pressure in each of them may be taken as sensibly constant. The pump M takes in air from C, compresses it adiabatically until its pressure becomes equal to the pressure in A, and then, the valve v being opened, delivers it into A. The indicator diagram for this action on the part of the pump is the diagram *fdae* in fig. 32. While this is going on, the same quantity of hot air from A is admitted to the cylinder N, the valve u is then closed, and the air is allowed to expand adiabatically in N until its pressure falls to the pressure in the cold chamber C. During the back stroke of N this air is discharged into C. The operation of N is shown by the indicator diagram *ebcf* in fig. 32. The area

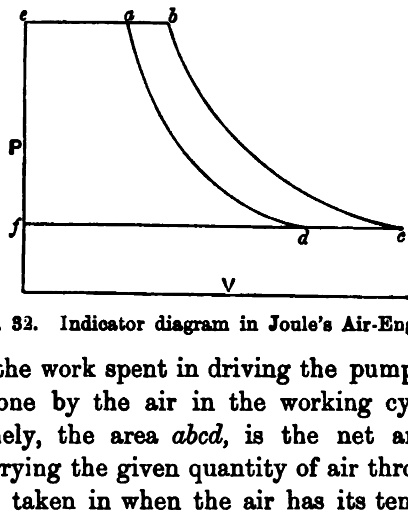


FIG. 32. Indicator diagram in Joule's Air-Engine.

fdae measures the work spent in driving the pump; the area *ebcf* is the work done by the air in the working cylinder N. The difference, namely, the area *abcd*, is the net amount of work obtained by carrying the given quantity of air through a complete cycle. Heat is taken in when the air has its temperature raised on entering the hot chamber A. Since this happens at a pressure which is sensibly constant,

$$Q_A = K_p (\tau_b - \tau_a),$$

where τ_b is τ_1, the temperature of A, and τ_a is the temperature reached by adiabatic compression in the pump. Similarly, the heat rejected

$$Q_C = K_p (\tau_c - \tau_d),$$

where $\tau_d = \tau_2$, the temperature of C, and τ_e is the temperature reached by adiabatic expansion in N. Since the expansion and compression both take place between the same terminal pressures, the ratio of expansion and compression is the same. Calling it r, we have

$$\frac{\tau_a}{\tau_d} = \frac{\tau_b}{\tau_e} = r^{\gamma-1}$$

(§ 39), and hence also

$$\frac{\tau_b}{\tau_a} = \frac{\tau_e}{\tau_d}, \text{ and } \frac{\tau_b - \tau_a}{\tau_a} = \frac{\tau_e - \tau_d}{\tau_d}.$$

Hence
$$\frac{Q_A}{Q_C} = \frac{\tau_a}{\tau_d} = \frac{\tau_b}{\tau_e},$$

and the efficiency

$$\frac{Q_A - Q_C}{Q_A} = \frac{\tau_a - \tau_d}{\tau_a} = \frac{\tau_b - \tau_e}{\tau_b}.$$

This is less than the efficiency of a perfect engine working between the same limits of temperature $\left(\frac{\tau_1 - \tau_2}{\tau_1}\right)$ because the heat is not taken in and rejected at the extreme temperatures.

The atmosphere may take the place of the chamber C: that is to say, instead of having a cold chamber, with circulating water to absorb the rejected heat, the engine may draw a fresh supply at each stroke from the atmosphere and discharge into the atmosphere the air which has been expanded adiabatically in N.

The entropy-temperature diagram for this cycle is drawn in fig. 33, where the letters refer to the same stages as in fig. 32. After adiabatic compression da, the air is heated in the hot chamber A and the curve ab for this process has the equation

$$\phi = \int_{\tau_a}^{\tau} \frac{K_p d\tau}{\tau} = K_p (\log_e \tau - \log_e \tau_a).$$

Then adiabatic expansion gives the line bc, and cd is another logarithmic curve for the rejection of heat to C by cooling under constant pressure. The ratio $\frac{\tau_a}{\tau_b}$ which is represented by $\frac{ea}{eb}$ in fig. 32 and by $\frac{ma}{nb}$ in fig. 33, shows the proportion which the volume of the pump M must bear to the

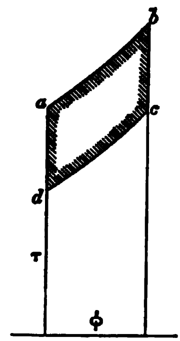


FIG. 33. Entropy-temperature diagram in Joule's Air-Engine.

volume of the working cylinder N. The need of a large pump would be a serious drawback in practice, for it would not only make the engine bulky but would cause a relatively large part of the net indicated work to be expended in overcoming friction within the engine itself.

In the original conception of this engine by Joule it was intended that the heat should reach the working air through the walls of the hot chamber, from an external source. But instead of this we may have combustion of fuel going on within the hot chamber itself, the combustion being kept up by the supply of fresh air which comes in through the compressing pump, and, of course, by supplying fuel either in a solid form from time to time through a hopper, or in a gaseous or liquid form. In other words, the engine may take the form of an *internal combustion* engine. Internal combustion engines, essentially of the Joule type, employing solid fuel have been used on a small scale, but by far the most important development of this type is the explosive gas-engine. Its cycle is substantially Joule's, considerably modified, however, by features which will be noticed in a later chapter.

This, however, is not the only reason why Joule's cycle is now interesting. In modern practice it has found application in the reversed form. Refrigerating machines in which air is the working substance are extensively used to keep the temperature of rooms on board ship below the freezing point, to allow frozen meat to be carried over seas, and such machines work, as we shall see immediately, by reversing the cycle suggested by Joule.

94. Reversal of the cycle in heat-engines: Refrigerating Machines or Heat-Pumps. By a refrigerating machine or heat-pump is meant a machine which will carry heat from a cold to a hotter body. This, as the second law of thermo-dynamics asserts, cannot be done by a self-acting process, but it can be done by the expenditure of mechanical work. Any heat-engine will serve as a heat-pump if it be forced to trace its indicator diagram backwards, so that the area of the diagram represents work spent on, instead of done by, the working substance. Heat is then taken in from the cold body and heat is rejected to the hot body.

Take for instance the Carnot cycle, using air as working substance (fig. 34), and let the cycle be performed in the order *dcba*, so that the area of the diagram is negative, and represents work spent upon the machine. In stage *dc*, which is isothermal expansion in contact with the cold body *C*, the gas takes in a quantity of heat from *C* equal to $c\tau_2 \log_e r$ (§ 40), and in stage *ba* it gives out to the hotter body *A* a quantity of heat equal to $c\tau_1 \log_e r$. There is no transfer of heat in stages *cb* and *ad*.

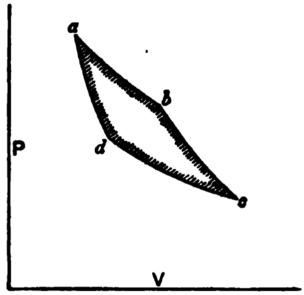


FIG. 34.

Thus *C*, the cold body, is constantly being drawn upon for heat and can therefore be maintained at a temperature lower than its surroundings. Suppose that such a machine were to be applied to the making of ice, then *C* might consist of a coil of pipe immersed in brine. The brine could in this way be kept by the action of the machine at a temperature below 32° F., and be used, in its turn, to extract heat by conduction from the water which is to be frozen. The "cooler" *A*, which is the relatively hot body, is kept at as low a temperature as possible by means of circulating water, which absorbs the heat rejected to *A* by the working air. This is substantially the process which is used in actual ice-making machines, except that the cycle of operations is not a reversed Carnot cycle, but more nearly a reversed Clausius cycle, and the working substance is a vaporisable liquid instead of air.

A machine using air as working substance and following Carnot's cycle would be exceedingly bulky. Its size would be considerably reduced if a regenerator, as in Stirling's engine, were resorted to in place of the two adiabatic stages of the Carnot cycle. Refrigerating machines of this kind, using air as working substance, with a regenerator, were introduced by Dr A. C. Kirk and were at one time considerably used[1]. The working air was completely enclosed, which allowed it to be in a compressed state throughout, so that even its lowest pressure

[1] See Kirk, On the Mechanical Production of Cold, *Min. Proc. Inst. C. E.* Vol. XXXVII., 1874. Also Lectures on Heat and its Mechanical Applications, *Inst. C. E.* 1884.

was much above that of the atmosphere. This made a greater mass of air pass through the cycle in each revolution of the machine, and hence increased the performance of a machine of given size.

This type of refrigerating machine has not survived, and those machines which now use air as working substance follow the reversed Joule cycle as described below in § 97.

95. Vapour Compression Refrigerating Machines. In most modern refrigerating machines, however, the working substance, instead of being air, consists of a liquid and its vapour, and the action proceeds by alternate evaporation under a low pressure and condensation under a relatively high pressure. A liquid must be chosen which evaporates at the lower extreme of temperature under a pressure which is not so low as to make the bulk of the engine excessive. Sulphuric ether was one of the earliest liquids to be used in this way, but ether machines were inconveniently bulky and could not be used to produce intense cold, for the pressure of that vapour is only about 1·3 lbs. per square inch at 4° F. and to make it evaporate at any temperature nearly as low as this would require the cylinder to be excessively large in proportion to the performance. This would not only make the machine clumsy and costly but would involve much waste of power in mechanical friction. The tendency of the air outside to leak into the machine is another practical objection to the use of so low a pressure. The liquids now used are carbonic acid, ammonia, and sulphurous acid. Of these ammonia is the most common. With ammonia it is easy to reach as low a limit of temperature as is required in any of the usual industrial applications of cold: the pressures are fairly but not excessively high, and the apparatus is compact.

Engines of this type are usually arranged to act as follows, in a cycle which is almost exactly the reverse of the Clausius cycle (§ 79). The organs, which are shown diagrammatically in fig. 35, are (1) a compressing cylinder, (2) a cold body C which serves as boiler for the volatile working fluid and allows heat to pass into the working fluid from the water or other substance that is to be made cold, and (3) a cooler A such as a coil of pipe surrounded by circulating water, in which the working fluid is condensed under pressure. The steps of the cycle are shown by the indicator

diagram in the same figure; *dc* is the forward stroke, during which the cylinder is taking in vapour from *C* at the uniform pressure corresponding to the lower limit of temperature τ_2. Compression of the vapour occurs during *cb*, which is the first part of the back stroke, and during which the valves leading to both chambers are shut. This continues till the pressure in the cylinder becomes equal to the pressure in *A*.

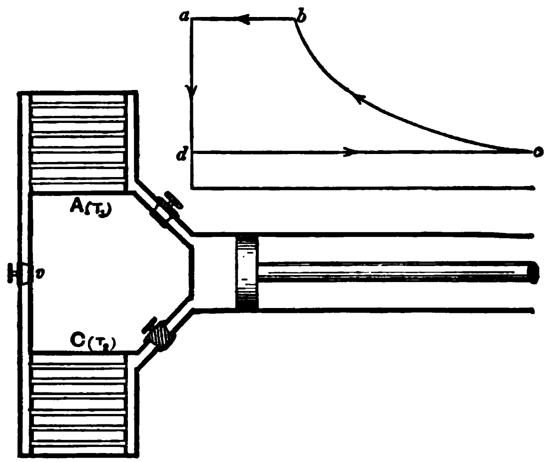


FIG. 35. Refrigerating Machine using the vapour of a liquid.

Next, the communication with *A* is opened and the back stroke is completed, the working substance passing into *A* and being condensed there (*ba*).

To complete the cycle, the same quantity of the substance is allowed to pass through the valve *v* directly from *A* to *C* (*ad*).

This last step in the process is not reversible, but it is a simpler way of completing the cycle than to complete it reversibly by letting the fluid do work in an expansion cylinder in passing from *A* to *C*, and the amount of work which would be saved if that were done is inconsiderable.

The operation of such machines may be represented by an entropy-temperature diagram like that of fig. 25, taking the lines in the reverse of the former order. If the evaporation were complete, a line such as *af* in that figure would represent the process of evaporation, during which heat is being taken in from the body to be cooled. More generally however evaporation is

incomplete; what is taken into the cylinder and compressed is a mixture of vapour with some of the unevaporated liquid. This reduces the superheating which compression would otherwise cause, and may even prevent superheating entirely, provided enough liquid be present in the mixture. Thus, in fig. 25, if *ad* represents the incomplete evaporation of the mixture which is taken into the cylinder, the adiabatic process of compression *dc* will bring the substance to a dry saturated state. On the other hand, if the original condition of the mixture is represented by *as*, adiabatic compression will superheat it to some extent, though not so much as if the evaporation had been complete before compression. In Dr Linde's form of ammonia compression machines superheating is prevented by using a wet mixture. The entropy diagram of the process is substantially like *adcb* of fig. 25. In other ammonia machines superheating occurs, and in some its effects are reduced by using a water-jacket to cool the cylinder.

Taking a cycle such as *adcb* of fig. 25, drawn for the particular substance and the particular temperatures used, it will be seen that the area under *ad* is the gross amount of heat taken up during evaporation of the working substance: but in passing from the condenser A to the refrigerator C the substance conveys with it an amount of heat equal to the area under the line *ba*. The net amount of refrigeration is the first of these areas *minus* the second. A substance in which the latent heat of the vapour is large compared with the specific heat of the liquid will cause the net refrigerating effect to approximate to the gross effect: in other words, in such a substance the reversed Clausius cycle will not differ much from the Carnot cycle which would correspond to the ideally efficient refrigerating process. Ammonia is from this point of view the most efficient of the substances which are now used in refrigerating machines. With carbonic acid, especially when the upper temperature approaches the critical point of the gas, the area under *ba* forms a relatively large deduction from the gross refrigerating effect.

96. Coefficient of Performance of Refrigerating Machines. The ratio

$$\frac{\text{Heat extracted from the cold body}}{\text{Work expended}}$$

may be taken as a coefficient of performance in estimating the merit of a refrigerating machine from the thermodynamic point of view. When the limits of temperature τ_1 and τ_2 are assigned it is easy to show by a slight variation of the argument used in § 45 that no refrigerating machine can have a higher coefficient of performance than one which is reversible in Carnot's sense. For let a refrigerating machine S be driven by another R which is reversible and is used as a heat-engine in driving S. Then if S had a higher coefficient of performance than R it would take from the cold body more heat than R (working reversed) rejects to the cold body, and hence the double machine, though purely self-acting, would go on extracting heat from the cold body in violation of the Second Law. Reversibility, then, is the test of perfection in a refrigerating machine just as it is in a heat-engine.

When a reversible refrigerating machine takes in all its heat, namely Q_C, at τ_2 and rejects all, namely Q_A, at τ_1, $\frac{Q_C}{\tau_2} = \frac{Q_A}{\tau_1}$ and the coefficient of performance

$$\frac{Q_C}{W} = \frac{Q_C}{Q_A - Q_C} = \frac{\tau_2}{\tau_1 - \tau_2}.$$

Hence—and the inference is highly important in practice—the smaller the range of temperature is the better. To cool a large mass of any substance through a few degrees will require much less expenditure of energy than to cool one-tenth of the mass through ten times as many degrees, though the amount of heat extracted is the same in both cases. If we wish to cool a large quantity, say of water or of air, it is better to do it by the direct action of a refrigerating engine working through the desired range of temperature, than to cool a portion through a wider range and then let this mix with the rest. This is only another instance of a wide general principle, of which we have had examples before, that any mixture or contact of substances at different temperatures is thermodynamically wasteful because the interchange of heat between them is irreversible. An ice-making machine, for example, should have for its lower limit a temperature only so much lower than 32° F. as will allow heat to be conducted with sufficient rapidity to the working fluid from the water that is to be frozen.

97. Reversed Joule Engine: the Bell-Coleman Refrigerating Machine. This machine was briefly mentioned in

§ 93 as one which has been, and still is, largely employed to maintain a cold atmosphere in the frozen-meat chambers of ocean steamships. It acts by drawing in a small portion of the air of the chamber, compressing that and extracting as far as possible by means of a cooler the heat developed by compression, then expanding the air until its pressure falls to that of the chamber. Its temperature is then lower than the temperature of the chamber in consequence of the removal of heat which took place while it was compressed. The air thus chilled by expansion is returned to the chamber, and in this way the temperature of the chamber is kept down notwithstanding the heat which reaches it by conduction from outside. The chamber has a thick lining of poorly conducting matter in order to reduce as far as may be the work which has to be spent on refrigeration.

The sketch, fig. 36, shows the organs diagrammatically. *C* is

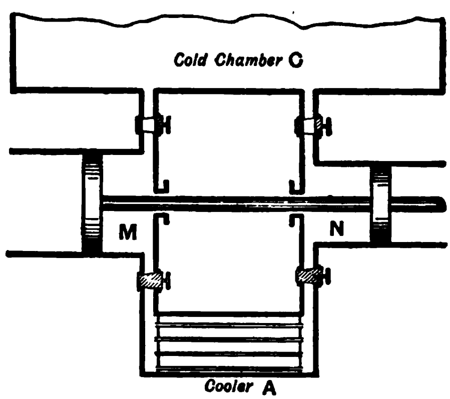


FIG. 36. Organs of the Bell-Coleman Refrigerating Machine.

part of the cold chamber, which is at or about atmospheric pressure, and *A* is the cooler, a set of pipes with circulating water. Compression takes place in *M* and expansion in *N*. *M* takes in air from *C* at temperature τ_2 during its out-stroke, and compresses that during part of its in-stroke till the pressure becomes equal to the pressure in *A*. These two operations are represented by the lines *fc* and *cb* in the indicator diagram, fig. 37. The compression *cb* has the effect of raising the temperature of the air above that of *A*. Consequently when the pump delivers the compressed air into *A*, by completing its return stroke (*bc*), which is the next

operation, the temperature of the air falls and a quantity of heat is rejected to A, namely

$$K_p(\tau_b - \tau_a),$$

where τ_b is the temperature reached by compressing, and τ_a is τ_1 the temperature of A. While this is going on, the cylinder N takes an equal quantity of air from A at τ_1 or τ_a, and expands it to the pressure of C: these operations are shown by the lines *ea*

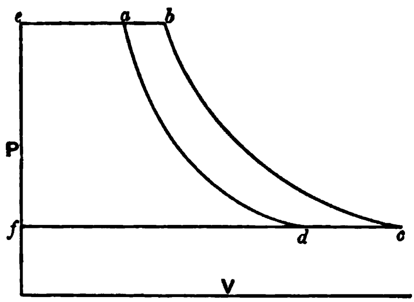


FIG. 87.

and *ad* in the indicator diagram. At the end of this expansion the temperature τ_d is lower than that of the cold chamber. Finally the chilled air is discharged into C during the return stroke of N, which is shown by the line *df* in the indicator diagram. The net amount of work expended is *badc*, *fcbe* being the indicator diagram of work spent upon the pump M and *eadf* being the diagram of work recovered in the expansion cylinder N. The net amount of heat taken from the cold chamber is $K_p(\tau_c - \tau_d)$. Assuming the processes *cb* and *ad* to be adiabatic, the ratio of expansion in N is equal to the ratio of compression in M, and hence $\frac{\tau_a}{\tau_d} = \frac{\tau_b}{\tau_c}$, as we have already seen in treating of the Joule cycle (§ 93) of which this is simply a reversal. Also $\frac{Q_A}{Q_C} = \frac{\tau_a}{\tau_d}$ and the coefficient of performance $\frac{Q_C}{Q_A - Q_C} = \frac{\tau_d}{\tau_a - \tau_d}$, a value less than $\frac{\tau_2}{\tau_1 - \tau_2}$ for the same reason that Joule's engine is less efficient than Carnot's.

In practice the compression in M is not adiabatic: by using a water-jacket, or by injecting water into the cylinder itself, the

compression may be made to follow a curve which lies between an adiabatic and an isothermal line. This has the thermodynamic advantage that some heat is extracted at a lower temperature than would be the case if compression were completed before the action of the cooler began. Or, to take another point of view, the net expenditure of work is reduced, since the compression curve from *c* rises less steeply than the adiabatic line *cb*.

A difficulty attending the use of machines of this type arises from the fact that the working substance is not *dry* air. It contains water-vapour in solution, as all air does except when specially dried, and when the temperature falls this tends to be condensed and even frozen. Difficulty has therefore in some cases been experienced from the clogging of valves and passages by snow or hoar-frost deposited by the working air. The air is generally saturated when it is cooled after compression (at the point *a* in the diagram), even when no injection-water is used to assist the cooling. It was to meet this objection that Mr Lightfoot introduced a form of the machine, in which the expansion was performed in two stages by means of a compound pair of expansion cylinders. In the first the temperature of the air was reduced to only about 35° F. At this temperature the greater part of the water-vapour was deposited as water, which was drained away, and the air then went on to the second cylinder, where it completed its expansion with very little further deposit of water. In Mr Coleman's form of the machine the compressed air after giving up heat through tubes to water in the ordinary cooler, is further cooled by passing through pipes which are exposed to the action of chilled air from the chamber, and is thus forced to give up its suspended moisture before it is allowed to expand[1]. This arrangement of "drying pipes" is still employed by Messrs Haslam in their construction of the Coleman apparatus, but other makers of air-refrigerating machines are now content with a mechanical separation of the air from any water which has been deposited in the cooling which precedes admission to the expansion cylinder. Provided the air

[1] For particulars of the construction and performance of these machines see Coleman, *Min. Proc. Inst. C. E.* Vol. LXVIII., 1882, p. 146; Lightfoot, *Proc. Inst. Mech. Eng.* 1881, p. 105, and 1886, p. 201. For a discussion of Refrigerating machines generally, see the Author's Howard Lectures on the Mechanical Production of Cold, *Jour. Society of Arts*, 1897; also the papers of Dr Kirk referred to above.

entering that cylinder is merely saturated and does not carry with it water in a state of mechanical suspension the deposit of snow is not so great as to be seriously troublesome.

The actual coefficient of performance of a machine of this class is much less than that of a machine using for working substance a vaporisable liquid such as ammonia. This is partly due to the relatively great waste of power, through friction, in air machines, and partly due to the practical necessity of using, in them, a much wider range of temperature than the range through which refrigeration is to be carried on. To keep the dimensions of the machine within reasonable bounds, the air is cooled by expansion to a temperature much lower than that of the cold chamber, and is heated by compression to a temperature considerably higher than that of the cooling water. When the working substance is a liquid which is being alternately vaporised and condensed the heat is much more easily got into and out of it. The efficiency of a vapour machine can be made to approach more closely to the ideal of a perfect refrigerator, and the coefficient of performance of a machine using ammonia is found in practice to be about five times that of an air machine.

98. The Reversed Heat-Engine as a Warming Machine. It was pointed out by Lord Kelvin in 1852 that the reversed heat-engine cycle might serve not only as a means of cooling but as a means of warming[1]. Let it be required for instance to raise and keep the temperature of a room above the temperature of the surrounding air. A machine of the Bell-Coleman type may take in air from the atmosphere, expand it so as to lower the temperature somewhat, and allow the temperature to rise again by conduction from external air. Then let it compress the air so as to restore it to atmospheric pressure. The temperature of the air will be thereby raised above the temperature of its surroundings, and it may then be discharged into the room which is to be warmed. The effect is, that by expending some mechanical work a quantity of heat is transferred from the cold atmosphere to the warmer room,—a quantity which may be far greater than the thermal equivalent of the work spent in driving the machine. For if the machine were reversible the

[1] *Proc. of the Phil. Soc. of Glasgow*, Vol. III. p. 269, or *Collected Papers*, Vol. I. p. 515.

heat rejected to the room A, namely Q_A, would be to the heat extracted from the atmosphere, namely Q_C, as τ_1 is to τ_2, and

$$\frac{Q_A}{W}=\frac{Q_A}{Q_A-Q_C}=\frac{\tau_1}{\tau_1-\tau_2}$$

where W is the work expended, expressed in thermal units. When the range of temperature is small Q_A may be many times greater than W, that is to say, a very large amount of heating through a small range may be achieved with but little expenditure of mechanical work.

The importance of the suggestion lies in the fact that the necessary power may be obtained, by means of a heat-engine, with a smaller supply of heat than would be required to effect the warming directly, provided the range of temperature of the warming be less than the range through which the heat-engine works in generating the required power. Burning fuel to warm a room by a few degrees is a wasteful way to utilise heat, even if all the heat of combustion be conceived to pass into the air of the room. The high-temperature heat produced in the combustion of coal or gas could warm a much larger volume of air to the same extent if it were applied to drive an efficient heat-engine, which in its turn drove a reversed heat-engine or warming machine to pump up heat through a short range of temperature from the diffused store of heat which is contained in the atmosphere or in the ocean. This is because a heat-engine can be arranged to take advantage of the high temperature at which heat is produced in the burning of fuel, whereas any direct communication of this high-temperature heat to a comparatively cool body, such as the air of a room, is thermodynamically bad. It is interesting, and may some day be useful, to recognise that even the most economical of the usual methods employed to heat buildings, with all their advantages in respect of simplicity and absence of mechanism, are in the thermodynamic sense spendthrift modes of treating fuel.

99. Heat-Engines employing more than one working substance: Steam and Ether Engines. So far as general thermodynamic principles are concerned the choice of working substance either in a heat-engine or a refrigerating machine is indifferent. The same efficiency is given by one substance as by another provided the character and range of the cycle be the same.

But the consideration that the pressure must be neither excessively high nor excessively low often determines whether one or another working substance is to be preferred. Vaporisable liquids have the advantage over air or any other permanent gas that heat can be more readily communicated to and extracted from them; but any such liquid has a comparatively limited range of temperature within which it is practicable for it to work. The efficiency of the steam-engine is, as we saw in § 68, largely conditioned by the fact that the upper limit of temperature cannot well exceed or even reach 400° Fah. This prevents full advantage being taken of the high-temperature heat which is generated in the combustion of fuel in boiler furnaces; and in this respect an air-engine has the superiority that in it a much higher temperature can be reached, since in a gas the connection of pressure with temperature is arbitrary. On the other hand a more volatile liquid than water would be even less suitable for use at high temperatures, unless indeed a large amount of superheating were employed. Going to the other end of the range, it will be seen by reference to the table of pressure and temperature for steam that a steam-engine is not well fitted to take full benefit of the low temperature which may be reached when there is condensing water at hand. A more volatile liquid would do this better, because its vapour could be expanded to the bottom of the range of temperature without making the pressure fall inconveniently low. With steam complete expansion would be useless, because in the last stages of the expansion the pressure would barely suffice to move the piston against its own frictional resistance; and therefore the *indicated* work which would be saved by completing the expansion would contribute nothing to the output of the engine.

For this reason it has been proposed to use what is called a "binary" heat-engine, that is, an engine with two working fluids, one to work through the upper part of the range, and another—a more volatile fluid—to work through the lower part. The less volatile fluid, namely water, after being evaporated in a boiler and after doing work in its cylinder, is condensed by passing through tubes in a vessel containing the more volatile fluid, to which it rejects heat. The more volatile fluid is thereby evaporated and does work in another cylinder, after which it is passed into a surface-condenser supplied with cold circulating water. A binary engine using ether as the more volatile fluid was introduced by

Du Tremblay about 1850[1], and the type has more than once been revived on a small scale[2].

The same principle of binary action has also been applied in refrigerating processes when very intense cold is to be produced for such purposes as the liquefaction of the permanent gases.

100. Transmission of Power by Compressed Air. A brief reference may be made in passing to the process, used on a large scale in Paris and elsewhere, of distributing power from central stations by compressing air there and conveying the compressed air through pipes to the places where it is to be used in driving engines, which are generally of the piston and cylinder type.

Imagine the compression to be performed exceedingly slowly, in a conducting cylinder, so that the air within may lose heat by conduction to the atmosphere as fast as heat is generated by

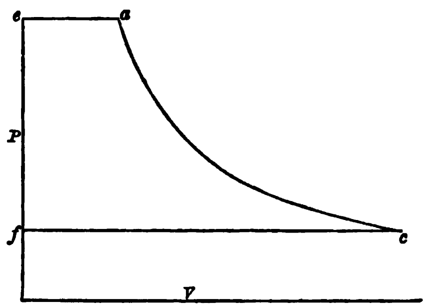


FIG. 38.

compression; the process will in that case be isothermal, at the temperature of the atmosphere. Imagine further that the compressed air is distributed without change of temperature, and that the process of expansion in the consumer's engine is also indefinitely slow and consequently isothermal. In that case (if we neglect the losses caused by friction in the pipes) there would be no waste of power in the whole process of transmission. The indicator diagram would be the same, per lb. of air, in the compressing engine as in the consumer's engine, namely *fcae* (fig. 38) in one and *eacf* in the other, *ac* being an isothermal line.

Imagine, on the other hand, that the compression and expansion

[1] See *Min. Proc. Inst. C. E.*, Vol. XVIII. p. 233. Also Rankine's *Steam-Engine*, p. 444.

[2] See *Min. Proc. Inst. C. E.*, Vol. CXII., 1893, pp. 481, 482.

are both adiabatic—a state of things which would be approximated to if they were performed very quickly. Then the diagram of the compression is *fcbe* (fig. 39) and that of the consumer's engine is *eadf* (fig. 40), *cb* and *ad* being adiabatic lines. The change of volume of the compressed air from *eb* to *ea* occurs through its cooling in the distributing pipes, from the temperature produced by adiabatic compression down to the temperature of the atmosphere. Superposing the diagrams as in fig. 41 and sketching an isothermal line between *a* and *c* (both of which are points at atmospheric temperature) we see that the use of adiabatic com-

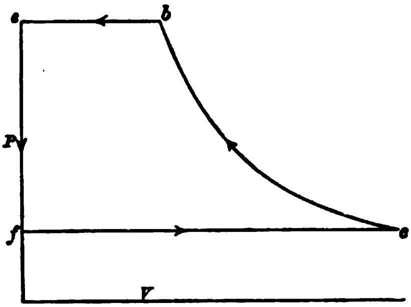


FIG. 39.

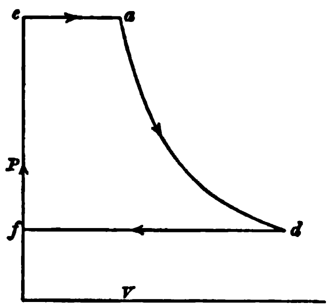


FIG. 40.

pression involves a waste of power which is measured by the area *cba*, while the use of adiabatic expansion by the consumer involves a further waste measured by *acd*.

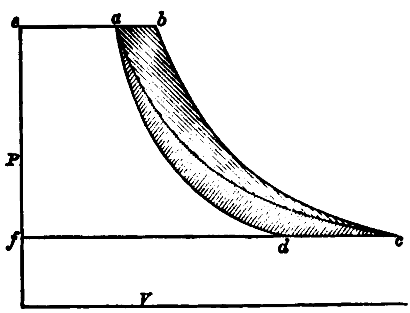


FIG. 41.

When the expansion and compression are both adiabatic, the efficiency of the process is readily found from the fact that a horizontal line drawn at any pressure to meet the curves *ad* and *bc* is divided by *ad* into segments the ratio of which is constant

and equal to the ratio of *ea* to *ab* or of *fd* to *dc*. Hence the efficiency, which is the ratio of the area of the expansion diagram to that of the compression diagram is $\frac{ea}{eb}$ or $\frac{fd}{fc}$. This may be expressed as $\frac{\tau_d}{\tau_e}$, or $\frac{\tau_d}{\tau_a}$ since $\tau_a = \tau_e$. Hence the efficiency is $\left(\frac{P_d}{P_a}\right)^{\frac{\gamma-1}{\gamma}}$. As an example, say that the air is compressed from 1 to 4 atmospheres: the efficiency is then $\frac{1}{4^{0.29}}$ or 0·67. When the expansion and compression are isothermal the efficiency, so far as these processes are concerned, is unity.

In practice the compression cannot be made strictly isothermal for want of time. The temperature of the air is prevented as far as possible from rising during compression by injecting water into the compressing cylinder, and in this way the curve which would be $PV=$ const. if isothermal and $PV^{1.4}=$ const. if adiabatic takes an intermediate position between *ca* and *cb* (as examination of the actual indicator diagram shows), and may be roughly expressed by the equation

$$PV^{1.2} = \text{const.}$$

Again, the waste of power in compression may be reduced by dividing the process into two or more stages (performed in two or more successive cylinders) and cooling the air between one stage and the next. In this way a stepped compression curve such as

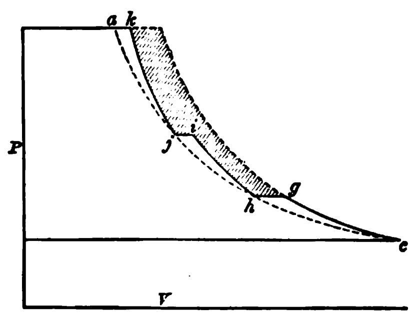


FIG. 42.

cghijk (fig. 42) can be obtained which approximates more nearly to the isothermal curve *ca*, and the loss is consequently reduced by the amount of the cross-hatched area. The saving so effected is considerable when air is highly compressed.

Similar devices may be used by the consumer to make the expansion curve in his engine approximate more nearly to the isothermal line: that is, he may inject water or use a compound engine, allowing the air time to take up enough heat to restore it more or less nearly to atmospheric temperature between one stage of expansion and the next. By these means the efficiency of the transmitting system as a whole (neglecting all losses due to friction in the distributing pipes, in the valves of the engines, &c.) may be made to approximate to unity.

There is, however, another point to be considered. If the temperature be allowed to fall materially during expansion the same difficulties present themselves as were referred to above in speaking of refrigerating machines: the expanding air tends to deposit dew or even snow. To prevent this the practice is often followed of passing the compressed air through a stove or "preheater" in order to raise its temperature just before it is allowed to expand, and so prevent the deposit of frozen moisture. When "preheaters" are used the extra heat which they supply is of course itself partly converted into work[1].

[1] On the subject of transmission of power by compressed air reference should be made to papers by Prof. Kennedy, *Brit. Assoc. Rep.* 1889, p. 448, and Prof. Nicolson, *Engineering*, July 7, 1893. See also Prof. Peabody's *Thermodynamics of the Steam-Engine*, Chap. xx.

CHAPTER V.

ACTUAL BEHAVIOUR OF STEAM IN THE CYLINDER.

101. Comparison of actual and ideal indicator diagrams. We have now to consider in what respects the action of steam in a real engine differs from the ideal action described in § 71 of Chapter III., where a hypothetical engine was considered in which the cycle of operation was as near an approximation to Carnot's as could be reached without the use of adiabatic compression. An engine imagined to work in the manner there described, and having an indicator diagram of the type shown in fig. 17, where the expansion is adiabatic and complete, forms a useful standard with which to compare real engines. Their efficiency is always less, for reasons which will be discussed in this chapter.

In the first place, the expansion in real engines is not (except in rare cases) complete: the steam at release has a pressure which is higher than the pressure in the condenser if the engine is a condensing engine, or higher than the pressure of the atmosphere if the engine is non-condensing. Reasons for this have been already indicated: complete expansion would increase unduly the bulk and weight of the engine; the work done by the steam in the last stages would add nothing to the net mechanical output for it would be used up in overcoming the friction of the piston; further, complete expansion would aggravate certain evils to be described later which arise from the cooling of the cylinder during expansion and exhaust. For these reasons it is practically desirable to cut off the toe of the ideal diagram sketched in fig. 17. The

effect which incompleteness in the expansion produces by itself on the efficiency of the ideal process has already been considered in reference to the indicator diagram, fig. 18, and to the entropy-temperature diagram, fig. 29 (§ 89).

Other features of difference are most conveniently noticed by comparing stage by stage the ideal diagram of fig. 18 with a diagram taken from a real engine. In the action to which figs. 17 and 18 refer it was assumed—(1) that the steam was supplied in the dry saturated state, and had during admission the full (uniform) pressure of the boiler P_1; (2) that there was no transfer

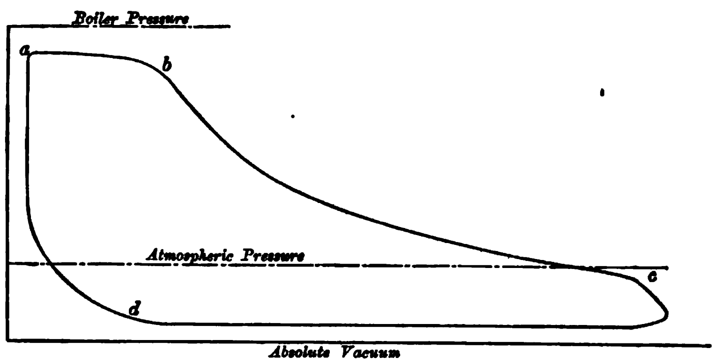


FIG. 43. Typical Indicator Diagram from a Condensing Steam-Engine.

of heat to or from the steam except in the boiler and in the condenser; (3) that after more or less complete expansion all the steam was discharged by the return stroke of the piston, during which the back pressure was the (uniform) pressure in the condenser P_2; (4) that the whole volume of the cylinder was swept through by the piston. It remains to be seen how far these assumptions are untrue in practice, and how the efficiency is affected in consequence.

The actual conditions of working differ from these in the following main respects, some of which are illustrated by the practical indicator diagram of fig. 43, which is taken from an actual engine.

102. Wire-drawing during Admission and Exhaust. Owing to the resistance of the ports and passages, and to the

inertia of the steam, the pressure within the cylinder is less than P_1 during admission and greater than P_2 during exhaust.

Moreover P_1 and P_2 are themselves not absolutely uniform, and P_2 is greater than the pressure of steam at the temperature of the condenser, on account of the presence of some air in the condenser. The presence of air is accounted for partly by its entering the boiler dissolved in the feed water, and partly by its leaking into the cylinder and other parts of the engine at times when the pressure within is less than the pressure of the atmosphere.

During admission the pressure of steam in the cylinder is less than the boiler pressure by an amount which often increases a little as the piston advances, on account of the increased velocity of the piston's motion and the consequently increased demand for steam. When the ports and passages offer much resistance the steam is expressively said to be "throttled" or "wire-drawn." Wire-drawing of steam is in fact a case of imperfectly resisted expansion (§ 50). The steam is dried by the process to a small extent, as was shown in § 77, and if initially dry it becomes superheated. In an indicator diagram wire-drawing causes the line of admission to lie below a line drawn at the boiler pressure, and generally to slope a little downwards. In fairly good practical instances the mean absolute pressure during admission is about nine-tenths of the pressure in the boiler. With a long steam-pipe or a badly designed valve the fall of pressure may be greater, and the effect is aggravated when the steam is allowed to become wet by leaving the steam-pipe bare or insufficiently covered, instead of having the pipe properly "lagged" with some material which is a poor conductor of heat. Even under the best conditions some of the steam is condensed on its way to the engine by loss of heat from the pipe. There is in general some additional water present in the steam through what is called "priming" on the part of the boiler, that is to say the delivery of steam in which particles of water are mechanically suspended. Whatever water is present, from either cause, may be more or less completely removed by the use of what is called a "separator," but usually the steam is to some extent wet when it enters the cylinder, notwithstanding the slight tendency which wire-drawing has to dry it. The separator is a vessel through which the steam passes on its way to the engine and in which the suspended particles of moisture settle, the accumulated

water being drained off from time to time. In many cases the steam is made to take such a course through the separator that the centrifugal action assists in causing the particles of water to be thrown off.

Again, during the exhaust the actual back-pressure exceeds the pressure in the condenser by an amount that depends on the freedom with which the steam makes its exit from the cylinder. In condensing engines with a good vacuum the back-pressure is often as much as 3 lbs. per square inch and even more, and in non-condensing engines it is 16 to 18 lbs. in place of the mere 14·7 lbs. or so which is the pressure of the atmosphere. The excess of back-pressure may be greatly increased by the presence of water in the cylinder. The effects of wire-drawing do not stop here. The valves open and close more or less slowly; the points of cut-off and release are therefore not absolutely sharp, and the diagram has rounded corners at b and c in place of the sharp angles which mark those events in fig. 18. For this reason release is allowed in practice to begin a little before the end of the forward stroke, hence the toe of the diagram takes a form like that shown in fig. 43. The sharpness of the cut-off, and to a less extent the sharpness of the release, depends greatly on the kind of valves and valve-gear used; valves of the Corliss type, for instance, which will be described in a later chapter, stop the admission of steam more suddenly than the ordinary slide valve does and therefore produce a diagram in which the events of the stroke are more sharply defined.

103. Clearance. When the piston is at either end of its stroke there is a small space left between it and the cylinder cover. This space, together with the volume of the passage or passages leading thence to the steam and exhaust valves, is called the *clearance*. It constitutes a volume through which the piston does not sweep, but which is nevertheless filled with steam when admission occurs, and the steam in the clearance forms a part of the whole steam which expands after the supply from the boiler is cut off. If AC be the volume swept through by the piston up to release, OA the volume of the clearance, and AB the volume swept through during admission, the apparent ratio of expansion is $\frac{AC}{AB}$, but the real ratio is $\frac{OA+AC}{OA+AB}$.

Clearance must obviously be taken account of in any calculation of curves of expansion. It is conveniently allowed for in indicator diagrams by shifting the line of no volume back through a distance corresponding to the clearance in the manner illustrated in fig. 44. In actual engines the volume of the clearance OA is usually from $\frac{1}{10}$ to $\frac{1}{50}$ of the volume of the cylinder. Its size depends largely on the kind of valve that is used. As a rule small engines have relatively more clearance than large ones.

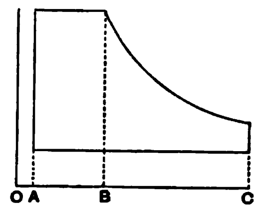


FIG. 44. Effect of Clearance.

104. Compression. Clearance affects the thermodynamic efficiency of the engine chiefly by altering the amount of steam that is consumed per stroke, and its influence depends materially on the extent to which the *compression* of part of the steam during the return stroke, referred to in § 71, is carried on. If there were no compression: if, in other words, the exhaust pipe leading to the condenser or to the atmosphere were left open throughout the whole of the back stroke, at the end of that stroke the clearance space would have nothing more in it than steam at a pressure equal to the back-pressure, and consequently at the next admission enough steam would have to be drawn from the boiler to bring up the pressure in the clearance as well as to fill the volume which is swept through by the piston up to the point of cut-off. With compression this cause of waste is more or less completely avoided. During the back stroke the process of exhaust is discontinued before the end as at d in fig. 43, and the steam remaining in the cylinder is compressed. The cushion of steam thus shut in finally occupies the volume of the clearance; and by a proper selection of the point at which compression begins the pressure of this cushion may be made to rise just up to the pressure at which steam is admitted when the valve opens. This may be called complete compression, and when it occurs the existence of clearance has no direct effect on the consumption of steam nor on the efficiency; for there is then simply a permanent cushion which is alternately expanded and compressed without net gain or loss of work, in addition to the working steam proper, which on admission fills the volume AB (fig. 44), and

which enters and leaves the cylinder in each stroke. But if compression be incomplete or absent there is, on the opening of the admission valve, an inrush of steam to fill up the clearance space. This increases the consumption to an extent which is only partly counterbalanced by the increased area of the diagram, and the result is that the efficiency is reduced. The action is, in fact, a case of unresisted expansion (§ 50), and consequently tends, so far as its direct effects go, to make the engine less than ever reversible. It is to be noted, however, that by any unresisted expansion of this kind the entering steam is dried to some extent, and this helps in a measure to counteract a cause of loss which will be described below. Incidentally, compression has the mechanical advantage that it obviates the shock which the admission of steam would otherwise produce, and increases the smoothness of running by giving the piston work to do while its velocity is being rapidly reduced—an action which receives the name of "cushioning."

The opening of the steam valve for admission being a somewhat gradual process, it generally begins before the back stroke is quite complete, in order that the valve may be widely enough open to let the steam in freely when the piston begins to move forwards. The valve is then said to have *lead*, and the effect is to produce what is called *pre-admission*. Pre-admission tends to increase the mechanical effect of cushioning which has just been referred to.

105. Cushion Steam and Cylinder Feed. In dealing with the influence of clearance, whether the compression be complete or incomplete or even altogether wanting, it is convenient to think of the working substance in the cylinder as made up of two parts, namely, (1) the part that has been shut up in the clearance from the previous stroke, and (2) the part that is freshly supplied from the boiler. For brevity we shall refer to these in what follows as (1) the cushion steam, and (2) the cylinder feed. During expansion the whole quantity of working substance in the cylinder is the sum of these two; during compression the cushion steam only is present. If the steam which leaves the engine is condensed and the condensed water weighed, its quantity forms a measure of the cylinder feed, from which the amount of steam passing through the cylinder per stroke may be deduced. But to this amount the cushion steam must be added when it is desired to know the whole weight of steam present in the cylinder.

106. Influence of the Cylinder Walls. Condensation and Re-evaporation in the Cylinder. Generally by far the most important element of difference between the action of a real engine and that of our hypothetical engine is that which was alluded to at the end of Chapter I., the difference, namely, which proceeds from the fact that the cylinder and piston are not non-conductors. As the steam fluctuates in temperature in the phases of admission, expansion and exhaust there is a complex give-and-take of heat between it and the metal it touches, and the effects of this, though not very conspicuous on the apparent form of the indicator diagram, have an enormous influence in reducing the efficiency by increasing the consumption of steam. Attention was drawn to this action by Mr D. K. Clark as early as 1855[1], and the results of his experiments on locomotives were confirmed and extended in 1860 by Mr Isherwood's trials of the engines of the United States steamer "Michigan[2]." Rankine in his classical work on the steam-engine notices the subject only very briefly, and takes no account of the action of the cylinder walls in his calculations. Its importance has now been established beyond dispute, notably, among early experiments, by those of Messrs Loring and Emery on the engines of certain revenue steamers of the United States[3], and by a protracted series of investigations carried out by M. Hallauer and other Alsatian engineers under the direction of Hirn[4], whose name should be specially associated with the rational analysis of engine tests, and who was one of the first to recognise the losses that result from condensation of steam on the surface of the cylinder. The evidence afforded by these experiments has been amply confirmed by an immense number of trials made on all kinds of engines and under every variety of working. In the next chapter some account will be given of how steam-engines are experimentally examined, and how, from the observed behaviour of the steam, we may deduce the exchanges of heat which occur between the steam and the cylinder throughout

[1] *Railway Machinery*, or art. STEAM-ENGINE, *Ency. Brit.*, 8th edition. See also *Min. Proc. Inst. C. E.*, vol. lxxii. p. 275.

[2] See Isherwood's *Experimental Researches in Steam Engineering*, Philadelphia, 1863. This important work describes a great number of experiments, undertaken at a time when engineers in general were but little alive to their value.

[3] An abstract of Messrs Loring and Emery's reports is given in *Engineering*, Vols. xix. and xxi., and in Mr Maw's *Recent Practice in Marine Engineering*.

[4] *Bull. Soc. Industr. de Mulhouse*, from 1877.

the stroke. The following is, in general terms, what experiments with actual engines show to take place.

When steam is admitted at the beginning of the stroke, it finds the metallic surfaces of the cylinder and piston chilled by having been exposed to low-pressure steam during the exhaust of the previous stroke. A portion of it is therefore at once condensed, and, as the piston advances, more and more of the chilled cylinder surface is exposed and more and more of the hot steam is condensed. At the end of the admission, when communication with the boiler is cut off, the cylinder consequently contains a film of water spread over the exposed surface, in addition to saturated steam. The boiler has therefore been drawn upon for a supply of steam greater than that which corresponds to the volume of the admission space. The importance of this will be obvious from the fact that the steam which is thus condensed during admission generally amounts to 30 and often even to 50 per cent. of the whole quantity that comes over from the boiler. Very rarely is it less than 25 per cent., and as much as 69 per cent. has been recorded in trials of a small engine[1].

Then, as expansion begins, more cold metal is uncovered, and some of the remaining steam is condensed upon it. There is in addition a further condensation which takes place in consequence of the work the steam is doing during expansion—a condensation which would be found even if the walls were perfect non-conductors and the process were strictly adiabatic. So far as these two actions are concerned, the mixture is getting wetter as it begins to expand. But the pressure of the steam now falls, and the layer of water which has been previously deposited begins to be re-evaporated as soon as the temperature of the expanding steam falls below that of the liquid layer. Hence, on the whole, the amount of water present increases during the earliest part of the expansion, but a stage is soon reached when the condensation which occurs on the newly exposed metal or throughout the steam as a whole is balanced by re-evaporation of older portions of the layer. The percentage of water present is then a maximum; and from this point onwards the mixture of steam and water present in the cylinder becomes more and more dried by re-evaporation of the layer.

[1] In papers by Col. English (*Proc. Inst. Mech. Eng.* Sept. 1887, Oct. 1889, May 1892), which describe experiments on this subject, and give the amounts of initial condensation which have been found in trials by a number of independent observers. In several cases the amount is over 60 per cent.

107. Re-evaporation continued during the exhaust. If the amount of initial condensation has been small this re-evaporation may be complete before release occurs. Very usually, however, there is still an undried layer at the end of the forward stroke, and the process of re-evaporation continues during the return stroke, while exhaust is taking place. In extreme cases, if the amount of initial condensation has been very great, the cylinder walls may fail to become quite dry even during the exhaust, and a residue of the layer of condensed water may either be carried over as water into the condenser, or, if the exhaust valves are not arranged so that it can be discharged, this unevaporated residue may gather in the clearance space, and in very bad cases may even require the drain-cocks to be left open to allow of its escape. When any water is retained in this way the initial condensation is enormously increased, for the hot steam then meets not only comparatively cold metal but comparatively cold water when it enters the cylinder. The latter causes much condensation, partly because of its high specific heat, and partly because it is brought into intimate mixture with the entering steam.

Apart, however, from this extreme case, whatever water is re-evaporated during expansion and exhaust takes heat from the metal of the cylinder, and so brings it into a state that makes condensation inevitable when steam is next admitted from the boiler. It is in fact the condensation of the layer and its re-evaporation, whether during expansion or during exhaust, that is the means of exchange of heat between the metal of the cylinder and the working substance. Mere contact with low-pressure steam during the later stages of expansion and during the exhaust stroke would cool the metal but little, for communication of heat between dry metal and any gaseous substance is slow even when the difference of temperature between them is large. The cooling which actually occurs is due mainly to the re-evaporation of the condensed water. Thus if an engine were set in action, after being heated beforehand to the boiler temperature, the cylinder would be only slightly cooled during the first exhaust stroke, and little condensation would occur during the next admission. But the metal would be more cooled in the subsequent expansion and exhaust, since it would part with heat in re-evaporating this water. In the third admission more still would be condensed, and

so on, until a permanent *régime* would be established in which condensation and re-evaporation were exactly balanced. The same permanent *régime* is reached when the engine starts cold.

However early the re-evaporation of the condensed film is completed it results in some chilling of the cylinder walls, leaving them to be re-heated by condensation of fresh steam in the next stroke. The evils of initial condensation are greater the later this re-evaporation is completed. If the steam in the condensed layer is all evaporated before the release but little further cooling of the metal will occur during the exhaust stroke: if water remains to be evaporated during exhaust the whole action of the sides is intensified. It is only in exceptionally favourable cases that the water condensed during admission is completely evaporated before release.

It is interesting to notice how important a part is played in the action of the cylinder walls by the wetness which results from the doing of work as the steam expands. Steam entering dry and becoming condensed during admission will give up to the metal an amount of heat equal (per lb.) to the latent heat L_1. On being re-evaporated at any lower pressure it will take up an amount of heat equal (per lb.) to the latent heat L_2 corresponding to that lower pressure. But in being cooled from the higher to the lower temperature it gives up an amount of heat equal to $h_1 - h_2$. On the whole so far as this condensation and re-evaporation of the same water is concerned the metal gains heat, since H_1 is greater than H_2 and consequently $L_1 + h_1 - h_2$ is greater than L_2. If therefore the metal were losing heat in no other way the process of condensation and re-evaporation could not go on. In actual cases, the metal does in general lose some heat by conduction to the outside; but the chief reason why the process goes on is because more water is evaporated than is initially condensed. This is because of the additional condensation which results from work being done as the steam expands.

108. Wetness of the working steam. Another reason why the process of condensation and re-evaporation is possible is that the steam generally brings in some water with it, which water is evaporated in the later stages of the stroke in addition to the water formed by condensation within the cylinder. Should the supply of steam be itself wet—apart from the wetness which

it acquires during admission on meeting the colder metal of the cylinder—the amount of water to be evaporated will be increased and consequently the action of the metal will be greater than if the steam were initially dry.

It follows from the reasoning of § 107 that for each lb. of water which is first condensed in the cylinder and then re-evaporated a net amount of heat equal to $L_1 + h_1 - h_2 - L_2$ must be in some way abstracted from the metal. This quantity, which may be written $H_1 - H_2$ or $0{\cdot}305\,(t_1 - t_2)$, is small compared with the latent heat L_2. If for instance the admission pressure is 90 lbs. and the exhaust pressure is 3 lbs. $t_1 - t_2$ is approximately 180°, and $0{\cdot}305\,(t_1 - t_2)$ is only about one-twentieth of L_2. Hence a comparatively small extra quantity of water, a small excess, that is to say, in the amount evaporated over the amount condensed, will be competent to account for a large amount of alternate condensation and re-evaporation. This extra quantity is supplied, when steam works expansively, by the condensation which is due to the work done during expansion, the condensation, namely, which would occur in an adiabatic process. But it may also be in part supplied by wetness in the supply. A small amount of such wetness will suffice to explain, so far as the heat balance is concerned, a low mean temperature on the part of the cylinder walls. Similarly a large effect may be produced by any loss of heat which the metal of the cylinder suffers by conduction and radiation from its outer surface[1].

At any stage in the expansion the whole quantity of water present may be regarded as consisting of two parts. The water which is formed in consequence of condensation during admission forms a film on the metallic surfaces. It may be conjectured, on the other hand, that the water which expansion itself produces rather takes the form of a mist of minute particles scattered throughout the whole volume. There are no experimental means of distinguishing between the two forms which the water may take, and the distinction would in any case have little interest. What we have to do with at all stages is simply a mixture, in varying proportion, of steam and water. Water in the form of mist—if anything of the kind is formed—would be much less easy

[1] On this subject reference should be made to an important investigation by Professors Callendar and Nicolson (*Min. Proc. Inst. C. E.* 1897–8) dealing with the whole question of condensation in the cylinder, and the cyclical variation of temperature in the cylinder walls.

to evaporate than water deposited upon the walls, and the fact that in some trials the fluid is apparently quite dry at the end of expansion rather indicates that when water is formed it collects wholly or mainly on the solid surfaces[1].

109. Graphic Representation, on the Indicator Diagram, of the Water present during Expansion. In testing engines, by methods which will be described in the next chapter, the amount of steam is measured which passes through the cylinder per stroke—that is, the quantity which we have called the "cylinder feed." The whole quantity of steam and water present during expansion is the cylinder feed *plus* the cushion steam. To estimate the amount of the cushion steam we take, on the indicator diagram, a point after compression has begun, after the exhaust valve has become completely closed, and note the pressure and the volume there, remembering that the true volume is the sum of the uncompleted portion of the stroke and the clearance. From this pressure and volume the quantity of the cushion steam is readily calculated, assuming that the steam is simply saturated and that no water is present when compression begins. As a rule, this assumption is probably correct: occasionally the cushion steam may be wet, which would make its amount greater, but in most cases the supposition that the steam is dry when compression begins may be accepted as involving at least no serious error. The total quantity of steam in the cylinder during expansion is next found by adding the amount of this cushion steam to the cylinder feed. A "saturation curve" can then be drawn on the indicator diagram to show the volume which this total quantity would fill if it were dry and saturated at each pressure reached during the expansion. An example is shown in the indicator diagram of fig. 45, where SS is the saturation curve. In drawing this line the axis of no volume is to be taken to the left of the diagram which the indicator traces, by a distance which represents the volume of the clearance. Then if a horizontal line ABS be drawn to intersect the expansion curve at any point B, AB is the

[1] In this connection reference should be made to the observations of Mr Bryan Donkin made by means of an apparatus which he has called a "revealer." See his papers "Sur les formes particulières prises par l'eau dans les cylindres de machines à vapeur" (*Revue Universelle des Mines*, 1893, p. 276, *Engineering*, June 30, 1893), and "Experiments on the Condensation of Steam" (*Min. Proc. Inst. C. E.* Vol. cxv., 1893).

actual volume which the expanding mixture filled at this pressure, AS is the volume it would have filled if dry and saturated; BS is the volume that is lost by wetness. Hence the proportion of water in the mixture is sensibly $\frac{BS}{AS}$, and the dryness q is $\frac{AB}{AS}$. Thus the proportion of water present at any stage of the expansion is determined and is shown in the diagram.

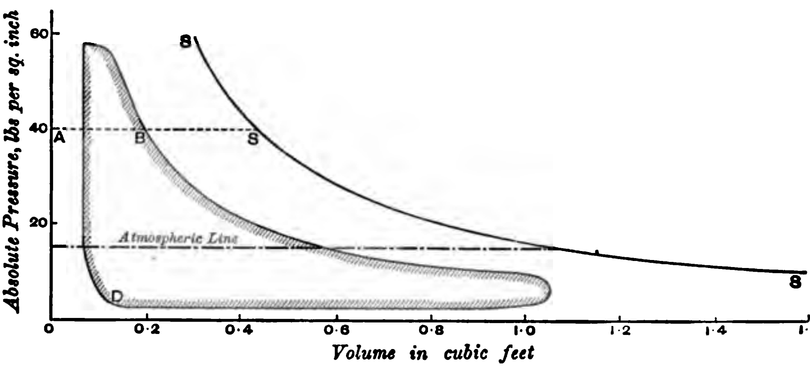


FIG. 45.

Fig. 45 relates to a real case—a trial, by the author, of a small engine of the marine type. The amount of cylinder feed per single stroke was 0·0404 lbs. The pressure at the point D was found to be 4 pounds per square inch, and the volume there was 0·12 cub. ft. Since the volume of 1lb. at that pressure is 90·4 cub. ft., it follows that the amount of cushion steam was 0·0013 lbs. This gives a total of 0·0417 lbs., for which the curve SS is drawn. By measuring values of $\frac{BS}{AS}$ at points along the curve it is found that the proportion of water in the mixture was 52 per cent. at cut-off, then increased to about 55 per cent. during the early stages of expansion, then became less, and finally sank to 37 per cent. just before release.

Again, knowing the wetness of the mixture at the point of cut-off we may draw an adiabatic line through that point using the equation $Pv^n = \text{constant}$ with a suitable value of n (see § 66). This curve will in general be found to lie a trifle above the actual expansion curve at first, but to cross it early and lie distinctly below it towards the end of expansion. This is because the metal continues for some time after cut-off to take heat from the working

fluid, but later gives up heat to it through the re-evaporation of the condensed film.

By comparing the adiabatic with the actual expansion curve it is possible to examine the give-and-take of heat between the metal and the working fluid. But this is more conveniently done after the entropy-temperature curve has been drawn, as will be presently described.

When tests of compound engines are in question it is useful to modify the construction shown in fig. 45 by separating the cylinder feed from the cushion steam, and drawing the diagram for the former. This allows a combined diagram for the several cylinders to be drawn, along with a single saturation curve. The reason is that the amount of cylinder feed is the same for both or all the cylinders, whereas the amount of cushion steam may be very different. An example of this construction will be given later in dealing with compound engine trials.

110. Use of the Entropy-Temperature diagram in exhibiting the behaviour of steam during expansion and the exchanges of heat between it and the cylinder walls. In the entropy-temperature diagram, fig. 46, let ab be drawn at the temperature which corresponds to the pressure at the point of cut-off, and let it be divided at c so that $\frac{ac}{cb}$ represents the proportion of dry steam to water in the total quantity of working fluid present in the cylinder. Similarly, at any lower temperatures reached during expansion let lines $a'b'$, $a''b''$ be divided at points c', c'' in the proportion of steam to water then present, making

$$\frac{a'c'}{a'b'} = \frac{AB}{AS}$$

at the corresponding pressure in the indicator diagram (fig. 45). In this way the curve $cc'c''$ is determined, which represents the real process of expansion, and this is readily compared with the ideal adiabatic process represented by the straight vertical line cg. Taking c'' as the point of release the diagram may be continued by drawing a constant-volume curve as described in § 89. In the first stages of expansion, namely from c to c' in the sketch, the proportion of water in the cylinder is increasing, and the heat abstracted by the cylinder walls from the steam is the area $cghc'$. From this point onwards the steam becomes drier,

and takes up heat from the metal, the whole amount recovered up to the point of release being the area $c'c''eh$. It will be seen that a diagram of this type is particularly well fitted to allow the transfer of heat between metal and fluid to be traced throughout all stages of the expansion, the heat given up or recovered in any

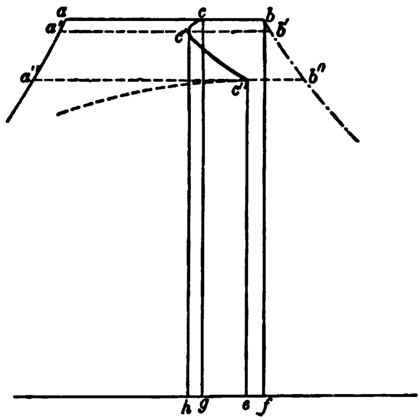


FIG. 46.

part of the process being equal to the area under the corresponding portion of the expansion curve $cc'c''$. When this curve slopes down to the left heat is passing from the steam to the metal; when it slopes down to the right the exchange is the other way. The heat abstracted from the steam during compression and admission is nearly equal to the area $fbcg$—nearly, but not exactly, because all the condensation in these stages does not occur at the pressure of cut-off. During compression condensation is going on at lower pressures because the temperature of the cushion steam—necessarily rising with the pressure—is being raised above the temperature to which the walls have been chilled during exhaust.

111. Thermodynamic Loss due to Initial Condensation. From a thermodynamic point of view all initial condensation of the steam is bad, for, however early the film of water be re-evaporated, this can take place only after its temperature has cooled below that of the boiler. The process consequently involves a misapplication of heat, since the substance, after parting with high-temperature heat, takes it up again at a temperature lower than the top of its range. This causes a loss of efficiency, and the loss is

greater the later in the stroke re-evaporation occurs. The heat that is drawn from the cylinder by re-evaporation of the condensed film becomes less and less effective for doing work as the end of the expansion is approached, and finally, whatever evaporation continues during the back stroke is an unmitigated source of waste. The heat it takes from the cylinder does no work[1]; its only effect, indeed, is to increase the back-pressure by augmenting the volume of steam to be expelled. A small amount of initial condensation reduces the efficiency of the engine but little; a large amount causes a much more than proportionally larger loss.

112. Action of a Steam-jacket. The action of the cylinder walls is increased by any loss of heat which the engine may suffer by radiation and conduction from its external surface. It has been pointed out in § 108 how any such loss tends to disturb the balance of heat in a cyclical process of condensation and re-evaporation, and consequently to promote condensation. More steam is initially condensed in a cylinder which is losing heat externally. The loss of efficiency due to the action of the cylinder walls will therefore be greater in an unprotected cylinder than in one which is well lagged or covered with non-conducting material. On the other hand, if the engine have a steam-jacket the deleterious action of the walls is reduced. The working substance is then on the whole gaining instead of losing heat by conduction during its passage through the cylinder. The jacket maintains a higher mean temperature on the inner surface of the cylinder, reduces condensation, and accelerates the process of re-evaporation, tending to make it occur while the temperature and pressure of the steam are still comparatively high. After the process of re-evaporation is complete the jacket cannot superheat the steam in the cylinder to any material extent, for conduction and radiation between dry steam and the metal of the cylinder are incompetent to cause any considerable exchange of heat. The earlier, therefore, that re-evaporation is complete the less is the metal chilled, and the less is the subsequent condensation. But after re-evaporation is completed the steam in the jacket continues to give heat to the metal during the remainder of the cycle, and so warms it to a temperature more

[1] Unless, of course, the cylinder in question is one of a compound series, and the steam that leaves it passes on to another cylinder to undergo further expansion there.

nearly equal to that of the boiler steam before the next admission takes place.

Thus a steam-jacket, though in itself a thermodynamically imperfect contrivance, inasmuch as it supplies heat to the working substance at temperatures lower than the top of the range, acts beneficially by counteracting, to some extent, the more serious misapplication of heat which occurs through the alternate cooling and heating of the cylinder walls. The heat which a jacket communicates to the working steam often increases the power of the engine to an extent far greater than corresponds to the extra supply of heat which the jacket itself requires. A jacket has the obvious drawback that it increases waste by external radiation, since it both enlarges the area of radiating surface and raises its temperature; notwithstanding this, however, many experiments have shown that the influence of a steam-jacket on the efficiency is good, especially in slow running engines and in engines where there is a large ratio of expansion in a single cylinder. This is to be ascribed to the fact that it reduces, though it does not entirely remove, the evils of initial condensation. To quote once more Watt's words, the jacket does good by helping to keep the cylinder as hot as the steam that enters it. To be effective, however, jackets must be well drained and kept full of "live" steam, instead of being, as many are, traps for condensed water or for air. The action is kept up by condensation of steam in the jacket itself. When the jacket is acting effectively the amount of steam which is condensed in it generally ranges from about 7 to 12 per cent. of the whole steam supply. The most economical treatment of the jacket-water is to allow it to drain directly back into the boiler. In some cases the activity of the jacket has been secured by letting all the steam supply pass through the jacket on its way to the cylinder, an arrangement which makes particular care necessary to prevent the water which is formed in the jacket from passing into the cylinder.

We shall refer presently to experiments which show the influence of steam-jackets on the efficiency of engines of various types. Meanwhile it may be said that in no trials has it appeared that a jacket has done harm: in other words, the saving of steam in the cylinder-feed brought about by the use of a jacket is always greater than the amount of steam which the jacket itself uses, and in many instances the net saving is as much as 10 or 20 per cent.

The best results are found in cases where, if the jacket were absent, the conditions are such as would give rise to much initial condensation. In engines which make a great number of strokes per minute the influence of the jacket is necessarily small[1].

The advantage of the jacket may be increased by making its temperature higher than that of the steam during admission to the cylinder. Re-evaporation of the condensed layer is further hastened, and after it is over the jacket gives up but little heat. Mr Bryan Donkin has obtained good results in experiments where the cylinder of a small engine was kept hot by gas flames, and it has been proposed to jacket engines with the hot gases from the furnace after these have passed through the boiler-flues[2].

113. Influence of Speed, Size, and Ratio of Expansion. It is interesting to notice, if only in general terms, the effects which the particular conditions of working in different engines may be expected to produce on the loss that occurs through the action of the cylinder walls. Initial condensation will be increased by anything that augments the range of temperature through which the inner surface of the cylinder fluctuates in each stroke, or that exposes a larger surface of metal to the action of a given quantity of steam, or that prolongs the contacts in which heat is exchanged. The influence of time is specially important; for the whole action depends on the rate at which heat is taken up and given up by the substance of the metal. The changes of temperature which the metal undergoes are in every case mainly superficial; the alternate heating and cooling of the inner surface initiates waves of high and low temperature in the iron whose effects are sensible only to a small depth; and the faster the alternate states succeed each other the more superficial are the effects[3]. In an engine making an indefinitely large number of strokes per minute the cylinder sides would behave like non-conductors and the action of the working substance would be adiabatic.

[1] See the Reports of the Inst. of Mechanical Engineers' Research Committee on the Value of the Steam-jacket. *Proc. Inst. Mech. Eng.*, 1889, 1892, 1894.

[2] For Mr Donkin's experiments see *Min. Proc. Inst. C. E.*, 1889, Vol. xcviii.

[3] The temperature of the cylinder walls has formed the subject of an interesting experimental study by Mr Bryan Donkin, who has examined the general gradient of temperature across the walls, both with and without steam in the jacket. See his papers, *Min. Proc. Inst. C. E.*, 1890 and 1891, also *Proc. Inst. Mech. Eng.*, 1895.

We may conclude, then, that in general an engine running at a high speed will have a higher thermodynamic efficiency than the same engine running at a low speed, all the other conditions of working being the same in both cases.

Again, as regards range of temperature, the influence of the cylinder walls will be greater (other things being equal) with high than with low pressure steam, and in condensing than in non-condensing engines.

In large engines the action of the walls will be less than in small engines, since the proportion of wall surface to cylinder volume is less. This conclusion agrees with the well-known fact that small engines do not readily achieve the economy that is reached in many larger forms.

Cylinder condensation is increased when the ratio of expansion is increased, all the other circumstances of working being left unaltered. The quantity of water formed by adiabatic condensation is then greater, and it is on this that the whole action mainly depends. Further, the metal is then brought into rather more prolonged contact with low-temperature steam. The volume of admission is reduced to a greater extent than the surface that is exposed to the entering steam, since that surface includes two constant quantities, the surface of the cylinder-cover and of the piston. For these and perhaps other reasons, we may expect that with an early cut-off the initial condensation will be relatively large; and this conclusion is amply borne out by experiment. An important result is that increase of expansion does not, beyond a certain limit, involve increase of thermodynamic efficiency; when that limit is passed the augmentation of waste through the action of the cylinder walls more than balances the increased economy to which, on general principles, expansion should give rise, and the result is a net loss. For this reason (as well as for the mechanical reason referred to in § 126, below) it is not wise in practice to carry expansion too far—not, in general, nearly so far as to be complete. With a given engine, boiler-pressure, and speed, a certain ratio of expansion will give maximum efficiency. But the conditions on which this maximum depends are probably too complex to admit of theoretical solution; the best ratio is a matter rather for experiment. It may even happen that an engine which is required to work at a specified power will give better results, in point of efficiency, with moderate steam-pressure

and moderate expansion, than with high steam-pressure and a very early cut-off.

114. Results of Experiments with various ratios of Expansion. The effect of increased expansion in augmenting the action of the sides and so reducing the efficiency, when carried beyond a certain moderate grade, was clearly shown by the American and Alsatian experiments alluded to above. The following figures (Table III.), relating to a single-cylinder Corliss engine, are reduced from one of Hallauer's papers[1] :—

TABLE III.

Ratio of Expansion.	Percentage of Water present.		Consumption of Steam per Hour per Indicated Horse Power. lbs.
	At End of Admission.	At End of Expansion.	
7·3	24·2	17·8	17·8
9·4	30·8	18·6	17·6
15·1	37·5	20·8	17·7

Here, in consequence of the amount of initial condensation increasing with increased expansion, a maximum of efficiency lies between the extreme grades of expansion to which the test extends, but the efficiency varies exceedingly little even through this wide range. In the American experiments the best results were obtained with even more moderate ratios of expansion. The

TABLE IV.

Ratio of Total Expansion.	Consumption of Steam per Hour per I. H. P. lbs.
4·2	21·2
5·7	20·0
7·0	20·3
9·2	20·7
16·8	25·1

[1] *Bull. Soc. Industr. de Mulhouse*, May 26, 1880.

compound engines of the United States revenue steamer "Bache," when tested with steam in the jacket of the large cylinder, with the boiler-pressure nearly uniform at 80 lb. by gauge, or 95 lb. per square inch absolute, and the speed not greatly varied, gave results which are shown in Table IV. Here the efficiency is very little affected by a large variation in the position of the cut-off, but when the ratio of expansion becomes excessive a distinct loss is incurred.

Again—to take a more recent instance, and one relating to a very different type of machine—trials made by the late Mr Willans with one of his high-speed compound non-condensing single-acting engines, using steam with an absolute initial pressure of 130 lbs., gave these results (Table V.)[1].

TABLE V. *Willans' Engine (Non-condensing): Effect of Varying the Expansion, the initial pressure and speed being constant.*

Ratio of Total Expansion.	Percentage of Water present at end of admission in high-pressure cylinder.	Consumption of Steam per Hour per I. H. P. lbs.
4	8·9	20·7
4·4	10·2	20·5
4·8	11·7	20·35
5·2	14·2	20·26
5·6	14·3	20·0
6	18·4	20·3
8	25·0	23·1

The initial condensation is comparatively small here, mainly in consequence of the exceptional speed (404 revolutions per minute), and for the same reason the economy in steam consumption is remarkably high for a small non-condensing engine. In another series of trials in which a compound engine of this type was worked with a condenser[2], and with steam at about 170 lbs. (absolute), Mr Willans found a slight increase in the steam consumption from 14·26 to 14·72 lbs. per hour per I. H. P. when the ratio of expansion was increased from 15½ to 20; at the same time the percentage of water present at cut-off on the high-pressure

[1] Willans on Non-Condensing Steam-Engine Trials. *Min. Proc. Inst. C. E.*, March 1888.

[2] Willans on Steam-Engine Trials. *Min. Proc. Inst. C. E.*, April 1893.

cylinder increased from 31 to 37. All these results agree in showing that the ratio of expansion may be varied through a large range with but little influence on the efficiency, because the gain that comes of making the expansion more complete is counterbalanced by the bad effects of increased initial condensation. The ratio of expansion which gives a maximum of efficiency is never sharply defined, and its value depends much on the initial steam-pressure and the particular features of the engine under trial.

115. Advantage of high speed. The advantage of high speed in making the action of an engine more nearly adiabatic has been demonstrated by experiment. Among the trials described by Mr Willans in his earlier paper are the following two sets made with one of his compound non-condensing engines, in the first set with an absolute admission pressure of 90 lbs. per square inch and 3·2 as the ratio of expansion; in the second set with 130 lbs. pressure and 4·8 as the ratio. In the three trials of each set the only condition varied was the speed.

TABLE VI. *Willans' Non-condensing Engine Trials: Influence of Speed.*

	I. Trials with Steam of 90 lbs. pressure.			II. Trials with Steam of 130 lbs. pressure.		
Speed: revolutions per minute.	401	211	122	405	216	131
Percentage of Water present at cut-off in the high-pressure cylinder.	5·0	12·6	20·2	11·7	19·1	29·7
Consumption of Steam per Hour per I. H. P. (lbs.)	24·2	25·3	27·0	20·3	21·3	23·7

The increase of steam consumption as the speed is reduced is considerable, and still more marked is the greater initial condensation. The same features are apparent in the trials quoted below (Table VII.) from an extensive series in Mr Willans' second paper; they relate to a condensing engine with an absolute admission pressure of 90 lbs. and a very moderate ratio of expansion (4·8).

TABLE VII. *Willans' Condensing Engine Trials: Influence of Speed.*

Speed: revolutions per minute.	401	301	198	116
Percentage of Water present at cut-off in the high-pressure cylinder.	8·9	12·2	17·9	20·9
Consumption of Steam per Hour per I. H. P. (lbs.)	17·3	17·6	18·9	20·0

116. Experiments on the value of the Steam-jacket. Abundant evidence of the advantage of the steam-jacket is given in Reports of a committee appointed by the Institution of Mechanical Engineers to enquire into the subject[1]. Individual figures vary widely, but it appears that the saving usually secured by jackets in condensing engines is something like 12 or 15 per cent. In non-condensing engines it is less. The following results of special trials with condensing engines are stated in the Report.

TABLE VIII. *Influence of Steam-jacket.*

Engine.	Total Steam per Hour per I. H. P. lbs.		Percentage less with Jackets.	Proportion of Jacket feed to total consumption per cent.
	Without Jackets.	With Jackets.		
Two-cylinder compound[1].	18·2	16·6	9	7
Two-cylinder compound[1].	24·7	20·0	19	6
Triple compound[1].	17·2	15·4	10	11
Triple compound[2].	16·4	13·6	17	
Two-cylinder compound[3].	21·1	19·5	7	12
Same engine run non-compound, the large cylinder only being used.	32·1	26·7	17	7
Small single-cylinder[4] engine.	39	29	25	7

[1] *Proceedings Inst. Mech. Eng.* 1889, 1892, and 1895.

[2] Prof. O. Reynolds' tests. For particulars see *Min. Proc. Inst. C. E.*, Vol. xcix. 1889.

[3] Prof. Unwin's tests: *Proc. Inst. Mech. Eng.* 1892, p. 460.

[4] Mr B. Donkin's tests: *Proc. Inst. Mech. Eng.* 1892, p. 464.

In several of these cases, notably in the last, it is remarkable how large a net saving of steam is secured by a comparatively small consumption in the jackets. In other trials of the same small engine, using an earlier cut-off, Mr Donkin found that 8 or 9 per cent. used in the jackets was capable of saving as much as 40 per cent. of the whole steam. In this instance there was excessive initial condensation when the jackets were out of use.

In compound engines the jackets are most effective when both or all of them are filled with steam at the boiler pressure. In Prof. Reynolds' triple engine trials, it was found that steam of the full boiler-pressure (200 lbs. per sq. inch) in all the jackets reduced the initial condensation in the second cylinder to about $\frac{1}{3}$ or $\frac{1}{4}$ of the amount that occurred without jackets, made the steam practically dry before the end of expansion in the second cylinder, and almost entirely prevented condensation in the third cylinder. Without steam in the jackets the second and third cylinder had been very wet, the proportion of water in them being about 40 per cent. of the whole. Indicator diagrams relating to these trials will be found in Chapter VII.

In Mr Donkin's experiments the temperature of the cylinder itself was observed at various points between the inner and outer surface, by means of thermometers inserted in small holes drilled in the metal. When the jackets were in use the mean temperature of the metal was almost equal to that of the steam on admission; when the jackets were not in use it was some fifty degrees lower. The temperature as shown by the thermometers was nearly uniform from inside to outside; for the periodic chilling of the innermost layer of metal by re-evaporation of condensed water was too superficial to be at all fully exhibited in this way.

117. Superheating. Superheating the steam before its admission reduces the amount of initial condensation, by lessening the quantity of steam needed to give up a specified amount of heat, and this in its turn lessens the subsequent cooling by re-evaporation. That it has a marked advantage in this respect was first experimentally demonstrated by Hirn, who found that the consumption of steam was reduced from 19·4 to 16·2 lbs. per horse-power-hour in a condensing engine by superheating the steam some 80° Fah. On general thermodynamic grounds superheating has a slight advantage (§ 86) because it allows a small

part of the whole heat supply to be taken in at a higher temperature than that of the boiler. But the indirect advantage is much more considerable. About the year 1860 superheating was frequently used in marine practice, but it was abandoned, mainly on account of difficulties in regard to lubrication. The importance of taking means to avoid or rather to reduce initial condensation was less generally understood in those days than it is now, and with the lubricants that are now used the old objections to superheating have little force.

So far as land engines are concerned, a revival in the use of superheated steam has now taken place. Many engines furnished with superheaters are in use, with results which make it probable that the practice will extend largely. Experiments made in 1892 by the Alsatian Association of Steam Users on a large number of engines furnished with superheaters showed that superheating effected a saving of coal to the extent of about 20 per cent. in cases where the superheater was simply placed in the boiler flue, so that it enabled what would otherwise be waste heat to be utilized, and about 12 per cent., on the average, in cases where the superheater was separately fired. Several of the engines tested were large, indicating 500 or 600 horse-power, and the superheating, which usually amounted to 60° or 80° Fah., appears to have been carried on without inconvenience. One of the trials, dealing with a triple-expansion Sulzer engine of 300 horse-power, records a consumption of 14·6 lbs. of steam per I. H. P.-hour without superheating, and 11·6 lbs. when the steam was superheated 100° Fah.

The following experiments made by Willans with and without

TABLE IX. *Willans' Engine with and without Superheating.*

Initial pressure, lbs. per sq. in. absolute.	45		35		25	
Temp. of Steam at admission (Fah.).	274°	305°	259°	299°	239°	289°
Amount of Superheating.	none	31°	none	40°	none	50°
Percentage of Water at cut-off.	21	17	24	19	25	15
Consumption of Steam per Hour per I. H. P.	26·7	24·6	23·9	26·4	30·0	26·4

superheating in a high-speed single-cylinder condensing engine cutting off steam at half stroke show that some advantage results even under conditions which are not such that much advantage can be expected.

Important trials of engines using superheated steam have been carried out in Germany by Prof. Schröter and Prof. Gutermuth. In one of Prof. Schröter's trials[1] a triple expansion factory engine indicating 1000 H. P. and supplied with steam at 100 lbs. per sq. inch was tested with saturated steam and with steam superheated to 420° Fah. (from a saturation temperature of 327° Fah.).

In the former case it used 13·2 lbs. per I. H. P.-hour, in the latter case only 12·0 lbs.[2] For this superheating, however, about 46 thermal units would be required, in excess of the heat taken up in forming saturated steam. If we assume the condensed water to be returned to the boiler at a temperature say of 100° Fah., the heat taken in per lb. would be 1114 units without superheating and 1160 units with superheating. The number of thermal units supplied, per I. H. P.-hour would be 14700 in the first case and 13920 in the other. The efficiencies, which are found by dividing 2545 by these numbers (§ 121), would be 0·173 and 0·183, showing a thermodynamic advantage of 5½ per cent. in favour of superheating.

Prof. Schröter's analysis of this test shows that the amount of the superheating was insufficient to prevent the steam from becoming somewhat wet during admission. At the point of cut-off in the first cylinder the steam was nearly, but not quite dry, and as expansion went on in this cylinder its wetness increased. The advantage, such as it is, of moderate superheating lies in reducing the losses which proceed from exchanges of heat between the steam and the cylinder walls. Although the steam retains its superheat until it reaches the engine, it at once falls to the temperature of saturation when it meets the cylinder walls. To keep it dry during admission requires an amount of superheating probably never less than 100° Fah. and often much more, the amount that is necessary depending on the ratio

[1] *Zeitschrift des Vereins deutscher Ingenieure*, Vol. xl. 1896.

[2] In reducing to British measure the results of tests stated in the metric system the number of kilogrammes per metric H. P.-hour has to be multiplied by 2·235 to bring it to lbs. per British H. P.-hour, the metric H. P. being 0·9863 of a British H. P.

of expansion in the cylinder and on the rate at which heat is lost from the external surfaces. To keep the steam dry during expansion a much higher degree of superheat would be needed. Superheating, in any moderate degree, may be regarded as a device for reducing the action of the cylinder walls by bringing the expansion curve of the indicator diagram nearer to the saturation curve (fig. 45) than it would otherwise come. In ordinary cases it barely makes the expansion curve come out so far as to reach the saturation curve even at cut-off, and as expansion proceeds the interval between the two increases. The ideal diagram sketched in fig. 25 is widely departed from in real superheated steam-engines, for the action of the cylinder walls in general keeps the whole process of expansion to the left of the saturation line *cf*. The working substance after being taken up the line *cr* before it reaches the engine, immediately comes down the line *cr* when it is admitted to the cylinder and before it begins to expand, giving up to the walls the heat (represented by the area under *cr*) which it has received in being superheated, and often going on to give up a part of its latent heat by becoming slightly wet. In cases where the superheat is high the steam comes only part of the way back from *r* towards *c* before it begins to expand.

118. Use of highly superheated steam. More striking advantages are found when steam is superheated 200 or 300 degrees above its temperature of saturation. In the Schmidt engine, which is specially designed for use with high temperature steam, the steam is superheated to about 660° Fah. by passing it through a coil of tubes in the uptake of the boiler furnace, and the consumption is the lowest on record in any form of steam-engine.

A two-cylinder compound condensing Schmidt engine of only 74 I. H. P. was tested by Prof. Schröter[1], with steam admitted at an absolute pressure of 170 lb. per sq. inch after being superheated to 662° Fah., or nearly 300 degrees above the temperature of saturation. The steam was found to be still considerably superheated at cut-off in the high-pressure cylinder, and to remain superheated during expansion, just becoming saturated at release, after expanding to about double its original volume. The consumption of

[1] *Zeitschrift des Vereins deutscher Ingenieure*, Vol. XXXIX. 1895.

steam, taking the mean of two trials, was 10·5 lbs. per I. H. P.-hour and 12·4 lbs. per brake H. P.-hour. The consumption of coal was 1·33 lbs. per I. H. P.-hour and 1·57 lbs. per brake H. P.-hour. Allowing for the heat taken in during the superheating, this figure corresponds to a thermodynamic efficiency of 0·20.

Even these remarkable results have been surpassed in trials of a larger engine by Prof. Gutermuth[1], and it has been found that the consumption of steam is in some cases barely 10 lbs. per I. H. P.-hour, corresponding to an efficiency of 0·21.

In trials by Mr Ripper[2] of a small non-condensing Schmidt engine, with various amounts of superheating, the most favourable case showed a consumption of 17·05 lbs. of steam with an admission pressure of 132 lbs. (absolute) and 326° of superheating. When the same engine was supplied with saturated steam, for which it is not well adapted, it consumed about 38 lbs. In this example the indicated H. P. was only 20 and the engine was non-compound. Under such conditions a consumption of 17 lbs. must be reckoned a very good performance. Taking account of the extra heat used in superheating, the consumption represents a supply of heat equivalent to that of 19½ lbs. of saturated steam, which is some twenty per cent. less than the consumption by an ordinary non-condensing engine of like size.

119. Advantage of Compound Expansion. The most important means in general use of preventing cylinder condensation from becoming excessive is the use of compound expansion. If the vessels were perfect non-conductors of heat it would be, from the thermodynamic point of view, a matter of indifference whether expansion was completed in a single vessel or divided between two or more, provided the passage of steam from one to the other was performed without introducing unresisted expansion. In practice, indeed, the transfer of steam from one cylinder to another during its expansion cannot be accomplished without more or less of wasteful drop in pressure. But the loss that this entails is more than counterbalanced by the gain that results from the reduced influence of the cylinder walls. Compound working acts beneficially by diminishing the range through which

[1] *Zeitschrift des Vereins deutscher Ingenieure*, XL. 1896, pp. 1390, 1419.

[2] *Min. Proc. Inst. C. E.*, Vol. CXXVIII. 1897, p. 60. Interesting examples will be found in the paper of entropy diagrams for an engine using superheated steam.

the temperature of any part of the cylinder-metal varies. For this reason the amount of steam initially condensed in the high-pressure cylinder of a compound engine is less than if admission were to take place at once into the low-pressure cylinder and the whole expansion were to be performed there. Further, the steam which is re-evaporated from the first cylinder during its exhaust does work in the second, and it is only the re-evaporation that occurs during the exhaust from the last cylinder that is absolutely wasteful. The exact advantage of this division of the whole range of temperature into two parts, or more than two, as compared with expansion between the same limits in a single cylinder, would scarcely admit of calculation; but it is easy to see in a general way that an advantage is to be anticipated, and experience bears out this conclusion. When a compound engine is tested first with compound expansion and then with the same grade of expansion in the large cylinder alone it is found that more steam is required per horse-power-hour in the second case.

Thus in the American Naval experiments the compound engine of the "Bache" when worked as a simple engine used 24 lbs. of steam per I. H. P.-hour, as compared with about 20 lbs. when the engine worked compound, with the same boiler pressure, the same total expansion, and steam in the jacket in both cases. Again, Professor Unwin's tests referred to in Table VIII. furnish another instance of the same thing: an engine taking 21 lbs. of steam per I. H. P.-hour when working compound required 32 lbs. when the large cylinder only was used, no jackets being then in action. With steam in the jackets the difference was rather less, for the jacket checked that excessive cylinder condensation which reduced the efficiency in the non-compound trials.

The general subject of compound expansion will be considered more particularly in a later chapter: at present we are concerned with the influence of compounding on efficiency. Experience shows that it is only by resorting to compound expansion that the economical advantages of high-pressure steam are to be secured. When high-pressure steam is used in a non-compound engine the waste due to initial condensation is excessive because of the great range of temperature through which the metallic surfaces fluctuate in every stroke. The necessity for compounding becomes greater with every increase of boiler pressure. So long as the initial

pressure is less than about 100 lbs. per square inch (absolute) it suffices to reduce the range of temperature into two parts by employing two-cylinder compound engines; with the considerably higher pressures now common in marine practice triple expansion is usual, and even quadruple expansion is occasionally employed.

The advantage of three cylinders is unquestionable; but it is doubtful whether—with the existing upper limit of pressure—four cylinders give any further saving sufficient to make up for the increased cost and complication of the machine.

120. Summary of Sources of Loss. The principal reasons have now been named which make the actual results of engine performance differ from the results which would be obtained if the steam conformed in every respect to the ideal cycle of § 71. The sources of loss may be summarised as follows:—

(1) Wire-drawing in admission and exhaust.

(2) Incomplete expansion before release.

(3) Incomplete compression of the cushion steam, through which the clearance becomes a cause of waste.

(4) The action of the metallic surfaces of the cylinder and piston, causing condensation during compression and admission, with re-evaporation during expansion and exhaust.

(5) Radiation and aerial convection of heat from steam pipe, valve chest and all hot parts of the engine, including evaporation and other escape of heat from condensed water in the hot well.

(6) Escape of the working fluid by leakage, and leakage of air into the condenser.

(7) Leakage of steam past the piston.

(8) In compound engines, additional wire-drawing or unresisted expansion in the transfer of steam from one cylinder to another.

If in drawing a comparison between the real engine and the ideal we take as the lower limit of temperature that of the cold water supplied to the condenser instead of the temperature of the hot well, we have a further item, namely, the loss that comes from the pressure in the condenser being higher than the pressure corresponding to this lower limit.

121. Methods of stating the performance of Steam-Engines. It remains to state a few of the best results that

have been obtained in trials of actual engines, from which the aggregate effect of these several sources of loss may be inferred. The methods used in making such tests will be described in the next chapter.

Statements of steam-engine performance may be put in a considerable variety of ways. Comparing the work done with the total heat which the working fluid takes in we may either calculate the thermodynamic efficiency (the work done divided by full mechanical equivalent of the heat taken in), or express the same idea in slightly different forms, such as by stating the number of thermal units used per indicated horse-power-hour. One horse-power being 33,000 foot-lbs. per minute, one horse-power-hour is 1,980,000 foot-lbs., the heat-equivalent of which is 2545 thermal units (taking J to be 778 foot-lbs.).

Thus the relation between these two modes of statement is given by the equation

$$\text{Efficiency} = \frac{2545}{\text{Number of thermal units used per horse-power-hour}}.$$

In reckoning the work done and the heat supplied it is very convenient to express these quantities per lb. of cylinder feed. A strict reckoning of the work done in the steam-engine cycle should allow for the work spent upon the working fluid in the feed-pump (and, in condensing engines, in the air-pump) an item which however is so small that account is rarely taken of it. Take for instance the case of an engine using say 16 lbs. of steam per horse-power-hour, the boiler pressure being 100 lbs. absolute. The feed-pump must return to the boiler 16 lbs. of water per I. H. P.-hour and the volume of this water is 0·26 cubic feet. The work spent upon it in transferring it from the condenser to the boiler is therefore $0·26 \times 100 \times 144$ or 3744 foot-lbs., a quantity almost negligibly small by the side of the 1,980,000 foot-lbs. which represents the work done by the steam.

The most usual plan followed by engineers in stating the results of trials is to give the number of lbs. of steam used per I. H. P.-hour. The supply of heat is nearly proportional to the supply of steam, and hence a knowledge of the latter gives at once a good general idea of the quality of the performance. There is very little change in the total heat of steam within the range of pressure at which boilers work in practice.

The most uncertain element in this form of statement is the dryness of the steam supply: if the boiler primes badly or if the steam is allowed to become very wet on its way to the engine, the quantity of heat which is supplied in each lb. of the cylinder feed may be seriously reduced. In an ill-designed or overworked boiler the amount of priming may be serious: under fairly good conditions it ought to be less, and is probably in general much less, than 5 per cent. of the whole supply. Unfortunately the amount of priming is difficult in any case to determine accurately by experiment.

When the boiler pressure, the feed temperature, and the dryness of the supply are known, it is practicable to treat the thermodynamic cycle as a whole and to calculate precisely how much heat is taken up by each lb. of steam. But even without making this calculation a mere statement of the number of lbs. of steam used per horse-power-hour is enough to allow a good judgment to be formed on the comparative results of different engine performances, so long as steam is supplied in the saturated state, and it has the advantage of putting results in a way that is easy to appreciate and remember.

The statement of results in lbs. of steam per I. H. P.-hour is however specially open to objection when we have to deal with superheated steam, since the whole supply of heat then depends on the extent to which superheating is carried. It would be convenient if the practice were generally followed of stating results in terms of the number of units of heat supplied per I. H. P.-hour[1]. In cases where n the number of lbs. of steam per I. H. P.-hour is stated, and where data are furnished for calculating Q the heat per lb. of steam (taking account when need be of priming or of superheating) the number of thermal units per I. H. P.-hour is nQ, and is equal to 2545 divided by the efficiency of the cycle.

122. Efficiency of boiler and furnace. "Duty." None of these modes of statement include the efficiency of the boiler and furnace. The performance of a boiler is most usually expressed by giving the number of pounds of water at a stated temperature that are converted into steam at a stated pressure by

[1] This practice is recommended by a committee of the Inst. of Civil Engineers. See Report for 1897.

the combustion of 1 lb. of coal. The temperature commonly chosen is 212° F., and the water is supposed to be evaporated under atmospheric pressure; the result may then be stated as so many pounds of water evaporated from and at 212° F. per lb. of coal. But the term "efficiency" may also be applied to a boiler and furnace (considered as one apparatus) to express the ratio of the heat that is utilized to the potential energy that is contained in the fuel. This ratio is, in good boilers, about 0·7. Thus, for example, 1 lb. of Welsh coal contains about 15,500 thermal units of potential energy, an amount which is equal to the heat of production (L) of about 16 lbs. of steam from and at 212°. In practice, however, 1 lb. of coal serves to evaporate only about 11 lbs. of water under these conditions, or about 9·5 lbs. when the feed-water enters, say, at 100° F. and the absolute pressure is 100 lbs. per square inch.

The efficiency of the engine multiplied by that of the furnace and boiler gives a number which expresses the ratio of the heat converted into work to the potential energy of the fuel,—a number which is, in other words, the efficiency of the system of engine, boiler, and furnace considered as a whole. Instead, however, of expressing this idea by the use of the term efficiency, engineers are rather in the habit of stating the performance of the complete system by giving the number of pounds of coal consumed per horse-power-hour. It must be borne in mind that this quantity depends on the performance of the boiler as much as on that of the engine, and that the difference in thermal value between one kind of coal and another makes it, at the best, a rough way of specifying efficiency. It is, however, an easy quantity to measure; and to most users of engines the amount of the coal-bill is a matter of greater interest than any results of thermodynamic analysis. Still another expression for engine, boiler and furnace performance taken jointly, similar to this last, is the now obsolete term "duty," which is the number of foot-pounds of work done for every 1 cwt. of coal consumed. Its relation to the pounds of coal per horse-power-hour is this—

$$\text{Duty} = \frac{112 \times 33000 \times 60}{\text{Number of lbs. of coal per I. H. P.-hour}}.$$

A good two-cylinder compound condensing engine of large size, supplied by good boilers, consumes about 2 lbs. of coal per horse-power-hour; its "duty" is then about 110 millions.

In the best examples of modern triple-expansion engines the consumption of coal is about $1\frac{1}{4}$ lbs. per horse-power-hour, making the "duty" about 166 millions.

123. Results of Trials: Non-Condensing Engines. The following are some representative results obtained in carefully performed trials of good engines.

In regard to non-condensing engines there are comparatively few records beyond the extensive series of tests by Willans on the special type of high-speed single-acting engine invented by him. Tests by Mr Emery of the single cylinder engines of the U. S. Steamer "Gallatin" included some non-condensing trials in which the consumption of steam was 25·9 lbs. per I. H. P.-hour with jackets in use and 30 lbs. without jackets[1]. The ratio of expansion was a little over 4, and the boiler pressure 70 lbs. by gauge. With the condenser in use, and the jacket, the consumption fell to 20·5 lbs. In tests by Mr J. W. Hill[2] three engines each with a single cylinder, with valves of the Corliss type, working at about 140 horse-power, without jackets, with a boiler pressure of 96 lbs., making 75 revolutions per minute, and cutting off steam at about $\frac{1}{6}$ of the stroke, required 25·9, 24·9 and 23·9 lbs. of steam per I. H. P.-hour respectively when worked non-condensing, and 20·9, 19·5 and 19·4 lbs. when worked condensing.

It appears then that under such conditions 24 lbs. of steam per I. H. P.-hour is a good performance for a non-condensing engine. It is interesting to compare this with the supply that would be required if the ideal conditions of § 71, including complete adiabatic expansion, were realized. In that case the amount of work obtainable from 1 lb. of steam is given (§ 79) by the equation

$$W = (\tau_1 - \tau_2)\left(1 + \frac{L_1}{\tau_1}\right) - \tau_2 \log_e \frac{\tau_1}{\tau_2}.$$

With a boiler pressure of 96 lbs. by gauge the temperature is 335° F. and τ_1 is 796. Since steam escaping to the atmosphere has the temperature 212° F. τ_2 is 673. By the table in the appendix $\frac{L_1}{\tau_1}$, which is $\phi_s - \phi_w$, is 1·106. We therefore have

$$\begin{aligned} W &= (796 - 673)(1 + 1{\cdot}106) - 673 \log_e 1{\cdot}183 \\ &= 259 - 113 = 146 \text{ thermal units.} \end{aligned}$$

[1] See Peabody's *Thermodynamics of the Steam-Engine*, p. 272.
[2] *Ibid.* p. 263.

To produce one horse-power-hour, which is equivalent to 2545 thermal units, we should therefore, under these ideal conditions, require $\frac{2545}{146}$ or 17·4 lbs. of steam, instead of the 24 lbs. or so which are actually required in these (exceedingly good) performances. In other words, the actual engine, at its best, succeeds in doing 73 per cent. of the work which it would be able to do if none of the sources of loss existed which were enumerated in § 120.

The same comparison may be made in this slightly different form. Since 24 lbs. of steam are used per H. P.-hour, each pound does an amount of work which is the equivalent of $\frac{2545}{24}$ or 106 thermal units, instead of the 146 units which it would do if it went through the ideal cycle.

Again, we may consider the thermodynamic cycle as a whole and compare it either with the ideal of § 71, or with the perfect cycle of Carnot. To do this something must be known or assumed as to the temperature of the feed-water, and we may take the most favourable possible case, namely that in which the escaping steam is collected at 212° F. as water and is returned to the boiler. In that event the heat taken in per lb. of steam would be

$$H_1 - h_2 = 1184 - 180 = 1004 \text{ thermal units.}$$

The work done per lb. of steam when 24 lbs. are used per horse-power-hour is

$$\frac{2545}{24} = 106 \text{ thermal units.}$$

The efficiency is therefore $\frac{106}{1004} = 0·1056$.

Compare this with the ideal of an engine having adiabatic expansion, but without adiabatic compression. Its efficiency would be $\frac{146}{1004}$ or 0·145. The actual engine, as we have seen, realises 73 per cent. of this.

Again, compare this with the efficiency of an engine following Carnot's cycle, namely

$$\frac{\tau_1 - \tau_2}{\tau_1} = \frac{796 - 673}{796} = 0·155.$$

Hence the actual engine's efficiency is 68 per cent. of that of a thermodynamically perfect engine working between the same limits of temperature, if we suppose the exhaust steam to be condensed at atmospheric pressure and returned to the boiler without loss of heat on the way. The standard of comparison here is of course different from the one just used; the calculation of W from equation (7) or (8) of § 79 may be said to afford a fairer means of contrasting what a steam-engine might do with what it does.

The important experiments carried out by Willans on high-speed engines of a special type, to which reference has been already made, included non-condensing trials both of single-cylinder and of compound engines[1]. With a small non-compound single-cylinder engine making 400 revolutions per minute the best result that is recorded was 26 lbs. of steam per indicated horse-power-hour, which was obtained when the mean absolute pressure during admission was 106 lbs. per square inch and the ratio of expansion was about $4\frac{1}{2}$. Higher efficiencies were reached in compound trials made at the same speed. With a boiler pressure of 100 lbs. per square inch above the atmosphere the compound engine took 23 lbs. of steam per horse-power-hour. At 135 lbs. per square inch in the boiler the consumption fell to 20·35 lbs. in one trial, and 20·75 in another. At 165 lbs. per square inch in the boiler it fell as low as 19·14 lbs. in two trials, and was under 19·2 lbs. in two others. Even these very remarkable results were surpassed in a few triple-expansion trials, when on raising the boiler pressure to 172 lbs. per square inch the consumption of steam per horse-power-hour was reduced to 18·5 lbs., in the mean of three independent and closely accordant measurements. All these figures refer to non-condensing engines.

Taking this last record the quantity W, calculated as above from the equation in § 79, is 183 thermal units. The actual engine got an amount of work out of each lb. of steam equivalent to $\frac{2545}{18 \cdot 5}$ or 138 thermal units, which is 75 per cent. of W. In most of the trials at lower pressures the proportion is about the same.

124. Results of Trials: Condensing Engines. A great number of good experiments on condensing engines have been

[1] *Min. Proc. Inst. C. E.* vols. xciii. and xcvi.

published: we must be content to give brief references only to a few.

Taking single-cylinder engines first, Mr Mair-Rumley's tests[1] of slow steam-jacketed engines, making 20 revolutions per minute and working at about 120 horse-power, show that with a pressure of 45 lbs. by gauge in the boiler the consumption of steam need not exceed 22 lbs. per I. H. P.-hour: in one case he records a consumption of 21·3 lbs. under these conditions. These figures correspond to a thermodynamic efficiency of 0·10. In Mr Hill's tests of faster Corliss engines the consumption was from 19·4 to 20·6 lbs. with a boiler pressure of 96 lbs. by gauge. Mr Willans' trials of one of his small high-speed single-cylinder engines, working with a condenser, gave 22·2 lbs. with a steam-chest pressure of 100 lbs. per square inch above the atmosphere, increasing to 30 lbs. as the pressure was lowered to about 5 lbs. per square inch.

Passing to compound engines, Mr Mair-Rumley's tests show that in three examples of steam-jacketed engines of about 130 horse-power, making about 25 revolutions per minute, the consumption of steam with a boiler pressure of 60 to 63 lbs. by gauge was 15·5, 15·1 and 14·8 lbs. per I. H. P.-hour. In these cases the steam was expanded about 14 times, and with this high expansion and slow speed the advantage of the jacket was very conspicuous. These figures correspond to a thermodynamic efficiency ranging from 0·14 to 0·15, and in view of the comparatively low pressure used must be ranked as exceptionally good results.

To compare these and other results of condensing tests with the performance "theoretically" possible we may take 100° F. as a standard lower limit of temperature; this is in fact about the usual temperature at which condensation takes place, and it is convenient to have a fixed standard. Values of W calculated by equation (7) of § 79 on the assumption that this is the lower limit are given for various absolute initial pressures in Table XI. (§ 125). They will be found useful in comparing the actual with the ideal performances of condensing engines.

In the best of the experimental results which have just been quoted the absolute boiler pressure was 76 lbs. per square inch, for which W is 274. The work actually done per lb. of steam was

[1] Described in two papers (J. G. Mair), *Min. Proc. Inst. C. E.*, Vols. lxx. and lxxix.

$\frac{2545}{14 \cdot 8}$ or 172 thermal units, which is not quite 63 per cent. of W. It should be noticed that this is a distinctly smaller fraction of the "theoretical" work than is realised in the best non-condensing trials. This, indeed, is a general characteristic of the performance of condensing engines: it rarely reaches even 60 per cent. of W, whereas 75 per cent. of W has been reached in non-condensing trials. The condensing steam-engine is less able to take full advantage of the lower part of its temperature range, for there is a greater difficulty in making the expansion approximately complete. Further, the greater range of temperature in condensing engines augments the prejudicial action of the cylinder walls.

The next example is a trial by Professor Unwin of a Worthington "High Duty" pumping-engine at West Middlesex Water Works[1]—a compound direct-acting steam-pump with no fly-wheel, steam-jacketed, and making about 17 double strokes per minute. Two tests were made in which the absolute boiler pressure was 90 and 75 lbs. respectively, the indicated horse-power being 296 and 256. The total consumption of steam was 17·4 and 17·7 lbs. per indicated horse-power-hour. For these pressures the values of W are 285 and 273½. The fraction of W actually obtained was 51 and 53 per cent. The thermodynamic efficiency was 0·13.

Tests of a compound pumping-engine of 250 horse-power (Leavitt) at the Boston Main Drainage Works[2], making 13 revolutions per minute, with an absolute boiler pressure of 114 lbs. per sq. inch, and steam-jacketed, showed a consumption of 13·9 lbs. of steam per indicated horse-power-hour in one trial and 14·2 lbs. in another. Taking 14·05 as a mean result, the amount of work got for 1 lb. of steam was 181 thermal units. This is 60 per cent. of W. The thermodynamic efficiency is 0·16. The consumption of coal was measured in the two trials, and was found to be at the rate of 1·33 and 1·35 lbs. per indicated horse-power-hour—figures which show that the performance of the boiler was on the same high level with that of the engine.

An important contribution to this subject will be found in the Reports of Marine Engine Trials made by a research committee

[1] *Engineering*, Dec. 1888.

[2] *Boston Soc. Civ. Eng.* 1885, or Peabody's *Thermodynamics of the Steam-Engine*, p. 293.

of the Institution of Mechanical Engineers under the chairmanship of Professor Kennedy[1]. These tests deal with large engines of modern construction and include several examples of the triple-expansion type. The performances are not equally good in all cases: the best of those recorded up to 1892 was made in a trial of the triple engines of the "Iona," of 650 H. P. The high-pressure cylinder only was jacketed. The absolute boiler pressure was 180 lbs. per sq. inch. It was found that 13·35 lbs. of steam and 1·46 lbs. of coal were used per indicated horse-power-hour. The work done per lb. of steam was therefore equivalent to 191 thermal units, which is 58 per cent. of W. The thermodynamic efficiency of this performance is nearly 0·17.

Professor Osborne Reynolds has published an account and discussion of trials made with the experimental triple-expansion engine of the Whitworth Engineering Laboratory at Owens College, which yielded some remarkably good results[2]. With steam in all the jackets, a boiler pressure (absolute) of 207 lbs. per sq. inch, these engines, running at about 300 revolutions per minute and indicating 72 horse-power, took 12·68 lbs. of steam per indicated horse-power-hour, and the consumption of coal was 1·33 lbs. This makes the work done per lb. of steam equal to 201 thermal units, which is 59 per cent. of W. The thermodynamic efficiency is 0·18. Small as these engines are a better performance has hardly ever been recorded.

Reference has already been made to the important series of trials made by Willans, which gave results almost equal to this last. In several of his triple-expansion condensing trials the recorded consumption of steam is below 13 lbs., and in one it is only 12·74 lbs. per indicated horse-power-hour. This was with an absolute steam-chest pressure of 185 lbs. per square inch (the boiler pressure, which must of course have been somewhat higher, is not stated). The engine made 375 revolutions per minute and had no jackets. Each pound of steam gave 200 thermal units of work, and if we take the steam-chest pressure as the upper limit in reckoning W, this is quite 60 per cent. of the "theoretically" possible quantity. If we were to take

[1] *Proc. Inst. Mech. Eng.* from 1889. The trials of the "Iona" were reported in April, 1891. For a summary of results of that and other trials see the Report of May, 1892.

[2] *Min. Proc. Inst. C. E.*, Dec. 1889.

the boiler pressure, as has been done in other cases, the realised percentage of *W* would still be close on 60. The thermodynamic efficiency is very nearly 0·18.

Throughout the two-cylinder compound condensing trials made by Willans the work done per lb. of steam was generally, at the highest speeds, from 50 to 55 per cent. of *W*. The least consumption of steam in them was 14·26 lbs. per indicated horse-power-hour, which was reached when the steam-chest pressure was approximately 175 lbs. Hence it corresponds to 54 per cent. of *W*.

Mr Donkin cites a number of exceptionally good results obtained in trials of the "Sulzer" engine by Professor Schröter and others[1]. Professor Schröter's trials show that a triple-expansion engine of about 200 horse-power, using steam with an absolute pressure of 171 lbs. per sq. inch, with jackets, consumed only 12·2 lbs. per indicated horse-power-hour exclusive of water condensed in the steam-pipe, and 12·56 lbs. when this condensed water was included. Taking the former figure the yield from each pound of steam was the mechanical equivalent of 208 thermal units, and since *W* was 326, this amounts to nearly 64 per cent., a proportion which is unusually large for a condensing trial. The thermodynamic efficiency, whether account be taken of the water condensed in the steam-pipe or not, is over 0·19.

In another trial by Professor Schröter of a triple-expansion Sulzer engine of 600 horse-power, the consumption of steam was 12·65 lbs. per indicated horse-power-hour, the absolute pressure of the steam being 160 lbs. per sq. inch. In two-cylinder compound engines of the same type the consumption is stated to be 14·3 lbs. per indicated horse-power-hour, as the mean of ten experiments made at pressures generally ranging from 100 to 105 lbs. (absolute). A single-cylinder Sulzer engine of 300 horse-power tested by Professor Linde consumed 19 lbs. of steam per horse-power-hour when the boiler pressure was 105 lbs. (absolute). This implies a yield equal to 45 per cent. of *W* and an efficiency of 0·12.

Table X. is a summary of the more important results which have been quoted for condensing-engine trials, using saturated steam. Results obtained with superheated steam have already been quoted in §§ 117 and 118.

[1] *The Engineer* and *Engineering*, Jan. 15, 1892.

TABLE X. *Results of Trials of Condensing Engines.*

Type of Engine.	Absolute boiler pressure lbs. per sq. inch.	Steam used per hour per I. H. P. lbs.	Percentage of W realized.	Thermodynamic Efficiency.
Single-cyl. beam pumping engine (Mair-Rumley).	59	21·3	46	0·11
Single-cyl. Corliss engine (Hill).	111	19·4	44	0·12
Single-cyl. Sulzer engine (Linde).	105	19·0	45	0·12
Two-cyl. compound beam pumping engine (Mair-Rumley).	76	14·8	63	0·15
Two-cyl. compound Worthington "High Duty" pumping engine (Unwin).	75 90	17·7 17·4	53 51	0·13 0·13
Two-cyl. compound pumping engine (Leavitt). (Mean of two trials.)	114	14·05	60	0·16
Two-cyl. compound high-speed single-acting engine (Willans).	180 (about)	14·26	54	0·16
Triple marine engine of S.S. "Iona" (Kennedy).	180	13·35	58	0·17
Triple experimental engine (Reynolds).	207	12·68	59	0·18
Triple high-speed single-acting engine (Willans).	190 (about)	12·74	60	0·18
Triple Sulzer engine (Schröter).	171	12·2	64	0·19

To these may be added an interesting trial by Mr M. Longridge[1] of a two-cylinder compound slow-running condensing-engine supplied with steam at a pressure exceptionally high for such an engine, namely, 148 lbs. per sq. inch (absolute). Both cylinders were jacketed with steam at the admission pressure, as was also the intermediate receiver. This with the slow running of the engines (34 revolutions per minute) served to keep the steam comparatively dry during expansion rates. Taking the second of

[1] Chief Engineer's Report of the Engine, Boiler, and Employers' Liability Insurance Co., 1895.

two trials the wetness in the high-pressure cylinder was only 25 per cent.) at cut-off, and 11 per cent. at release, although the cut-off occurred at about one-sixth of the stroke. In the low-pressure cylinder the wetness was 22 per cent. at cut-off and 12 per cent. at release. The indicator diagrams taken in this trial are shown in Fig. 47 for the high-pressure cylinder and Fig. 48

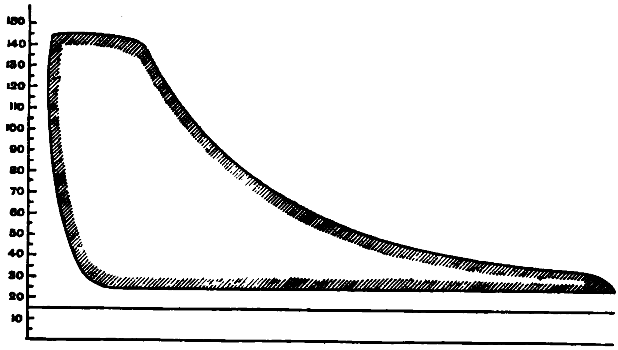


FIG. 47.

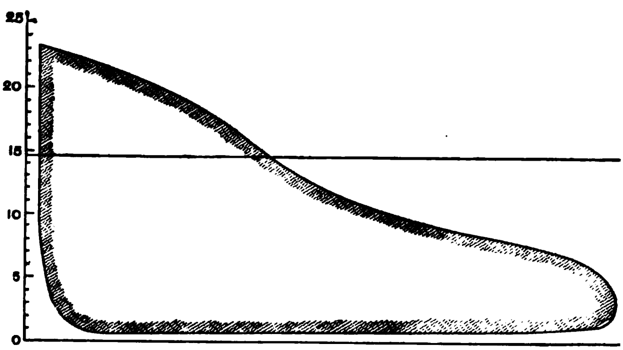


FIG. 48.

for the low-pressure cylinder. A large total ratio of expansion was secured by giving the second cylinder four times the volume of the first, and by cutting off steam early in the high-pressure stroke. The engine developed 221 I. H. P., and the consumption of steam, as measured by the condenser discharge together with the jacket drains, was 12·82 lbs. per I. H. P. hour. The number of thermal units used per I. H. P. hour was 14700, and the efficiency

was 0·17. The work done was the equivalent of 198 thermal units per lb. of steam, which is about 62 per cent. of W. Of the whole supply of steam the jackets took nearly 8 per cent. These results are nearly, though not quite, as good as those which have been reached in the best triple expansion engines. They show that in favourable cases an engine with two cylinders may do practically as well as an engine in which the complication of a third cylinder has been introduced.

§ 125. **Standards to be used for comparison with the results of tests.** In quoting the foregoing results of tests comparison has incidentally been made with certain ideal thermodynamic processes. It may be well to consider more particularly what are the various ideals which may be set up as standards in the criticism of the performance of real engines[1].

Taking the cycle as a whole, from hot well back to hot well through feed-pump, boiler, engine and condenser, we may find the efficiency in a real performance (or alternatively the heat units supplied per horse-power-hour) and compare that with the corresponding quantity in an ideal cycle.

The ideal may be the perfect cycle of Carnot or it may be the less exacting standard which is frequently described as the cycle of Clausius (§ 71), the cycle, namely, in which the substance suffers complete adiabatic expansion, but is returned to the boiler without having its temperature raised by adiabatic compression. Referring to the entropy diagram of Fig. 23, the cycle of Clausius is represented (for initially saturated steam) by the figure *abcda*; and for superheated steam by the figure *abcrsa* of Fig. 25.

Of these two ideals, the cycle of Clausius furnishes the more useful standard, as coming nearer to the conditions which must hold even in the very best steam-engine. Feed heating by adiabatic compression is never practised, and may fairly be said to be impracticable. In a real steam-engine, the more perfectly we can prevent incidental losses due to external conduction and radiation, to the action of the cylinder walls, and to irreversible movements of the steam such as are caused by wire-drawing or by imperfect expansion, the more nearly do we come to the Clausius

[1] For a further discussion of this subject reference may be made to a paper by Captain Sankey, *Min. Proc. Inst. C. E.*, Vol. cxxv. 1896, p. 182.

ideal. Moreover for superheated steam the Carnot cycle is inappropriate, for it fails to take account of the fact that the substance must take in heat below the top of the temperature range.

In the Clausius cycle, as in the Carnot cycle, the ideal efficiency is conditioned, for saturated steam, by the limits of temperature at which steam is produced on the one hand and condensed on the other. To calculate the Clausius efficiency for superheated steam we must, in addition, know the temperature of superheating.

Taking the case of saturated steam, if the object of the comparison is to see what efficiency could be reached by taking the fullest possible advantage of the available range of temperature, the Clausius cycle should be calculated with the boiler temperature as the upper limit, and the temperature at which condensing water is supplied as the lower limit. The real performance will fall short of this ideal, first, because the steam wastes heat on its way from the boiler to the engine; second, because it fails to utilize heat in the best possible way within the engine itself through the action of the cylinder walls, and through imperfectly resisted expansion; and third, because the condenser is imperfect, the pressure under which the steam is condensed being higher than would correspond to the temperature at which the condensing water is supplied. It is not possible strictly to separate these sources of loss. It is evident, however, that if we wish to avoid making the engine suffer for the faults of the steam-pipe a fairer value of the Clausius standard will be got by taking as upper limit the temperature of the (saturated) steam just before its admission to the engine, say at the engine stop-valve. Again, if we wish to avoid making the engine proper suffer for the faults of the condenser, a fairer lower limit will be the temperature within the condenser, or the temperature of the steam in the exhaust pipe. This will differ in different cases: as a rule it is not far from 100° Fah., and that temperature has been adopted in some of the preceding calculations of the work W ideally obtainable from the steam.

Table XI. has been calculated with this as the lower limit, assuming the Clausius cycle to be followed; and the values there stated have been used in Table X. for comparison with the actual result of trials.

TABLE XI. *Work theoretically obtainable from one lb. of Saturated Steam, with complete adiabatic expansion, assuming the lower limit of temperature to be* 100° *Fah.*

Absolute pressure of supply lbs. per sq. inch.	W Thermal Units.	Absolute pressure of supply lbs. per sq. inch.	W Thermal Units.
50	248	135	311
55	254	140	313
60	259½	145	315½
65	264½	150	318
70	269	155	320
75	273½	160	322
80	277½	165	324
85	281½	170	326
90	285	175	328
95	288½	180	329½
100	292	185	331
105	295	190	333
110	298	195	334
115	301	200	335½
120	303½	205	337
125	306	210	338½
130	308½	215	340

126. Mechanical Efficiency of the Engine. All these figures refer to the indicated horse-power,—to the work done upon the piston by the steam. But as the object of a steam-engine is to drive some other machine or machines it is important to recognise the distinction between the indicated work done in the cylinder and that quantity of work (always smaller) which the engine does against external resistance. Say that the engine is set to work against a brake, the "brake horse-power" or "effective horse-power" will be less than the indicated horse-power by an amount which represents the expenditure on friction of one kind and another in the mechanism of the engine itself. The ratio of the one to the other is the mechanical efficiency of the engine. In favourable cases the mechanical efficiency is about 0·85; in other words, some 15 per cent. of the indicated work is ineffective, being spent on friction within the engine. Occasionally the mechanical efficiency may approach 0·9; in general, however, it is a good deal less. When a steam-engine is directly employed to drive a dynamo, the comparison is often made between the electrical

output and the indicated work: in that case the efficiency of the dynamo is of course involved as well as that of the engine as a mechanism. Similarly in dealing with pumping-engines, the comparison is usually made between the indicated work and the work usefully applied by the pump—this latter being determined by the volume and pressure of the fluid delivered. Here again the mechanical efficiency of the pump and of the engine are both involved.

In the Worthington engine trials referred to above the output of the pump represented 84 to 85 per cent. of the indicated work done by the steam. In the Leavitt trials also it was 84 per cent. In Professor Reynolds's tests, when the engine was loaded by means of brakes, the brake horse-power was 82 per cent. of the indicated power in the most favourable instances.

With a given engine running at a given speed the work expended in driving the engine itself is usually a nearly constant quantity, whether much or little effective work is being taken off. Hence the mechanical efficiency is reduced when the load on the engine is lightened. We shall have occasion in the next chapter to revert to this subject and to describe means of finding and of stating the loss which the energy suffers in its transmission through the mechanism.

The consideration of mechanical efficiency furnishes a mechanical reason for not carrying the expansion of steam in an engine cylinder so far as to be complete—a reason which is entirely independent of the augmented condensation referred to in § 113. If the "toe" of the indicator diagram be extended to the limit of complete expansion, the latest portion of the stroke is ineffective, and worse than ineffective, so far as external work is concerned. Though the area of the indicator diagram is somewhat increased the work done outside is diminished. Loss of effective work begins as soon as the expansion is carried further than a limit at which the pressure on the piston is just able to overcome the friction within the engine. So far as brake horse-power is concerned it is a positive advantage to cut off the toe of the diagram by letting the steam escape from the cylinder as soon as this limit has been reached. For the same reason it is obviously very wasteful to make the expansion more than complete, in other words to bring the pressure below the pressure of discharge and form a loop at the toe of the diagram.

127. Curve of Expansion to be assumed in estimating the probable indicated horse-power of steam-engines.

Largely as the exchanges of heat between the working substance and the cylinder affect the consumption of steam, their influence on the form of the expansion curve is but slight. In practical cases the curve is never very different from a rectangular hyperbola. The simple supposition that the pressure during expansion varies inversely with the volume will answer sufficiently well in forming conjectural indicator diagrams for such a purpose as the estimation of the probable power to be exerted by an engine of given size, when the speed, the initial pressure, the back-pressure and the ratio of expansion are assigned.

If there were no clearance, if the full pressure of supply, p_1, were maintained during the admission, if the cut-off and release were perfectly sharp, if the expansion continued to the very end of the stroke, and if there were a uniform back pressure p_b, without compression, then the assumption that the curve of expansion may be treated as a common hyperbola would make the mean effective pressure equal to

$$\frac{p_1(1+\log_e r)}{r} - p_b,$$

where r is the ratio of expansion, namely, the ratio which the whole volume of the stroke bears to the volume that is swept through up to the point of cut-off. To show this, we may express the area of the indicator diagram as

$$p_1 v_1 + \int_{v_1}^{v_2} p\,dv - p_b v_2$$

where v_1 and v_2 are the volumes at cut-off and release respectively, so that $r = \frac{v_2}{v_1}$.

Since by assumption pv at any point in the expansion $= p_1 v_1$, the area is

$$p_1 v_1 \left(1 + \int_{v_1}^{v_2} \frac{dv}{v}\right) - p_b v_2$$

$$= p_1 v_1 (1 + \log_e r) - p_b v_2.$$

The mean effective pressure is found by dividing the area by v_2, when the formula given above is obtained. The same formula would be applicable to compound expansion, r being interpreted

as the final ratio, if the further condition were satisfied that there should be no loss of pressure during the transfer of the steam from one cylinder to the next.

In practice, of course, these conditions are not fulfilled, and the general result is to make the actual mean effective pressure p_m less, in a proportion which is sometimes stated by the use of a coefficient e, thus:

$$p_m = e\left\{\frac{p_1(1+\log_e r)}{r} - p_b\right\}.$$

The diagram factor, as Professor Unwin[1] has called e, is a number less than unity, which may be estimated as a matter of experience from the results given by other engines of like types, working under more or less similar conditions.

[1] *The Practical Engineer*, June 17, 1892. See also remarks by Mr C. H. Wingfield, *Min. Proc. Inst. C. E.*, Vol. cxiv. p. 83, and *Engineering*, Oct. 20, 1893.

CHAPTER VI.

THE TESTING OF STEAM-ENGINES.

128. The Indicator. In this chapter we have first to describe the ordinary process of taking indicator diagrams, whether for the purpose of finding the horse-power of an engine or of examining the action of the steam; then to speak of those further measurements that have to be made when the thermodynamic efficiency of the engine is under trial; and finally to mention a method of finding the brake horse-power which, by comparison with the indicated power, gives the mechanical efficiency. The indicator diagram, apart from its use in determining power, is invaluable as an index of what is going on within the cylinder. It shows the time and manner of the four events of the stroke, namely, the admission, cut-off, release and compression, which together make up what is called the "distribution" of the steam; it detects faults in the setting or in the working of the valves and suggests changes by which the distribution may be improved. When the information which it gives is supplemented by a knowledge of how much steam is passing through the cylinder per stroke a complete analysis of the action becomes possible; the wetness of the steam at any stage may then be determined, as well as the exchanges of heat that take place between the steam and the cylinder walls.

The indicator, invented by Watt and improved by M'Naught and by Richards, consists of a small steam cylinder, fitted with a piston which slides easily within it and is pressed down by a spiral spring of steel wire. The cylinder of the indicator is connected by a pipe below this piston to one or other end of the cylinder of the engine, so that steam from the engine cylinder has free access and

the piston of the indicator consequently rises and falls in response to the fluctuations of pressure which occur in the engine cylinder. The indicator piston actuates a pencil, which rises and falls with it and traces the diagram on a sheet of paper fixed to a drum that is caused to turn back and forth about its axis through a certain angle, in unison with the motion of the engine piston. In M'Naught's indicator the pencil was directly attached to the indicator piston, in Richards's the pencil is moved by means of a system of links so that it copies the motion of the piston on a magnified scale. This has the advantage that an equally large diagram is drawn with much less movement of the indicator piston, and errors which are caused by the piston's inertia are consequently reduced. In high-speed engines especially it is important to minimize the inertia of the indicator piston and the parts connected with it. In Richards's indicator the linkage employed to multiply the indicator piston's motion is an arrangement similar to the parallel motion which was introduced by Watt as a means of guiding the piston-rod in beam engines. In several recent forms of indicator lighter linkages are adopted, and other changes have been made with the object of fitting the instrument better for high-speed work. One of the best of these modified forms of Richards's indicator is that made by the Crosby Company, which is shown in figs. 49 and 50. The pressure of

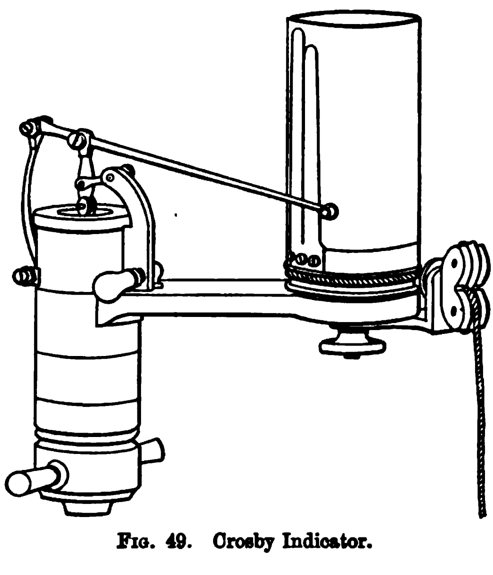


FIG. 49. Crosby Indicator.

steam in the engine cylinder raises the piston *F* (which is shown in section in fig. 50), compressing the spring above it and causing

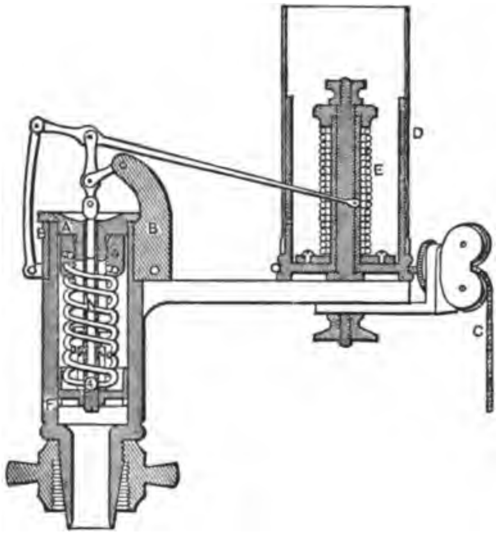


FIG. 50. Crosby Indicator. Sectional view.

the pencil to rise in a nearly straight line through a distance proportional, on a magnified scale, to the compression of the spring, and therefore to the pressure of the steam. At the same time the drum *D*, which carries the paper, receives motion through the cord *C* from the cross-head of the engine. Inside this drum there is a spiral spring *E* which becomes wound up when the cord is pulled, and serves to turn the drum in the reverse direction during the back stroke. The cap *A* of the indicator cylinder has holes in it which admit air freely to the top of the piston, and the piston has room to descend, extending its spring, when the pressure of the steam is less than that of the atmosphere. The spring is easily taken out and replaced by a more or less stiff one when higher or lower pressures have to be dealt with. Springs adapted to various ranges of pressure are supplied with the indicator and are marked with a number which states the pressure, in lbs. per square inch, which will raise the pencil through a distance of one inch on the paper. The accuracy of this number should be verified by testing the indicator under steam against a standard pressure-gauge, or (better) a mercury column. Tests with water-pressure are not

suitable, unless a proper allowance be made for the change of elasticity of the spring through change of temperature. The spring is stiffer (generally by two or three per cent.) when cold than at the temperature (about 212° F.) which it takes up when in use. The pressure of the steam should be raised slowly or by steps, and the test should be made with rising and also with falling pressure, to see that there is no material friction-error, which would show itself by causing the indicator pencil to stand higher during the fall than during the rise when the steam had the same pressure.

A tap, placed just below the indicator but not shown in the diagram, allows the communication with the cylinder to be closed at pleasure, and also puts the space below the indicator piston into communication with the atmosphere, thereby allowing the "atmospheric line" to be drawn on the diagram.

The pencil is withdrawn from the paper by turning back the piece *BB* which is separate from the rest of the indicator cylinder. The small handle shown in fig. 49 is provided for this purpose, and a stop behind the handle prevents the pressure of the pencil against the paper from exceeding a regulated amount.

Fig. 51 shows another arrangement of pencil gear and spring used in one of the forms of indicator made by Messrs Elliott Brothers. Here the spring is outside, away from the action of the steam, and its form is such that it is easily adjusted and easily removed and replaced.

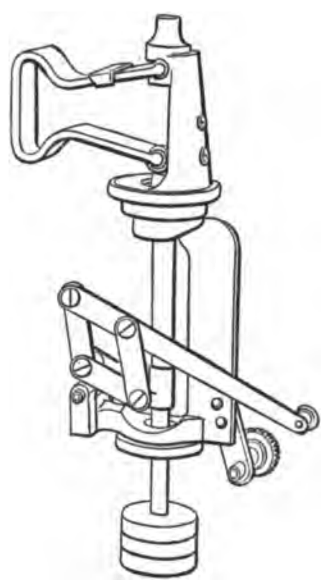


FIG. 51.

In the Wayne Indicator, also by Messrs Elliott (fig. 52), neither the steam cylinder and piston nor the paper drum, which are common features in other indicators, are used. The paper is carried on an aluminium cradle which slides back and forth, and the pencil is actuated by an oscillating disc-piston which turns about the axis of a cylindrical chamber to which the steam is admitted. The marking point is a

radial arm projecting from the oscillating axis of this disc-piston, and the cradle which carries the paper is so formed as to give

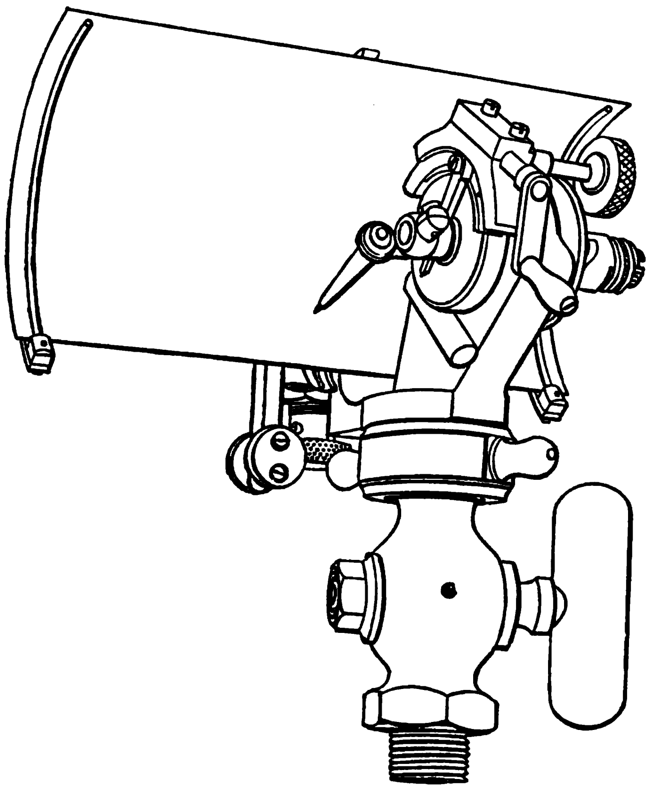


FIG. 52. Wayne Indicator.

the paper a cylindrical surface corresponding to the radius of the pencil point. The spring is at the other end of the oscillating axis, outside the steam chamber. The gear which appears in the diagram on the top of the steam chamber is a special attachment, which may be removed in the ordinary use of the indicator, and has the purpose of limiting by stops the motion of the pencil to a very small angle, so that any particular part of the diagram may be made the subject of separate examination. This form of indicator has great advantages in cases where the inertia of the ordinary pencil movement causes trouble.

129. Conditions of accurate working. To register correctly, an indicator must satisfy two conditions: (1) the motion of the piston must be proportional to the change of steam-pressure in the engine cylinder; and (2) the motion of the drum must be proportional to that of the engine piston.

The first of these conditions requires that the pipe which connects the indicator with the cylinder shall be short and of sufficient bore, and that it shall open in the cylinder at a place where the pressure in it will not be affected by the kinetic action of the inrushing steam. Frequently pipes are led from both ends of the cylinder to a central position where the indicator is set, so that diagrams may be taken from either end without shifting the instrument. This arrangement is convenient and shows the double action prettily; but except with small cylinders it makes the connecting pipes so long as to give rise to serious errors. In large engines it is therefore not admissible: a pair of indicators should rather be used, each fixed with the shortest possible connecting pipe, or the diagrams should be taken successively from the two ends of the cylinder with a single instrument set first at one end and then at the other. The general effect of an insufficiently free connexion between the indicator and the engine cylinder is to make the diagram too small.

The first condition of correct working is also invalidated to some extent by the friction of the indicator piston, of the joints in the linkage, and of the pencil on the paper. The piston must slide very freely; nothing of the nature of packing is permissible, and any steam that leaks past it must have a free exit through the cover. The pencil pressure must not exceed the minimum which is necessary for clear marking. By careful use of a well-made instrument the error due to friction in the piston and connected parts may be kept so small as not to be serious. Another source of disturbance is the inertia of these parts, which tends to set them into oscillation whenever the indicator piston suffers a comparatively sudden displacement. These oscillations, superposed upon the legitimate motions of the piston, give a wavy outline to parts of the diagram, especially when the speed is great and when the last-named source of error (the friction) is small. When they appear on the diagram a continuous curve may be sketched by hand midway between the crests and hollows of the undulations. To keep the oscillations within reasonable compass

in high-speed work a stiff spring must be used and an indicator with light parts has to be selected. Still another possible source of error is backlash through too great looseness in the joints. Finally, to secure accuracy in the pencil's movement, the strain of the spring must be kept well within the limit of elasticity, so that the strain may be as nearly as possible proportional to the steam-pressure.

With regard to the motion of the drum, it is, in the first place, necessary to have a reducing mechanism which will give a sufficiently accurate copy, on a small scale, of the engine piston's stroke. Many contrivances are used for this purpose; in some a rigorous geometrical solution of the problem is aimed at—as for instance by adapting some form of pantagraph—but it is not unusual to find that the multiplicity of joints in such mechanisms gives rise, through backlash, to greater errors than would occur in simpler forms of gear designed to produce no more than a close approximation. Of these simpler gears a very usual form is that shown in fig. 53.

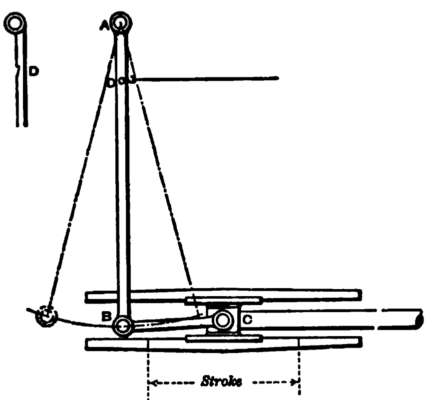


FIG. 53.

AB is a long pendulum rod, pivoted on a fixed centre at *A*. *BC* is a short link connecting *B* with the cross-head *C*. The indicator drum receives its motion from any suitable point *D* on the rod *AB* by a cord which is led over pulleys if necessary. A very convenient arrangement, shown by the separate sketch at the side, is to have a notch in the rod *AB* at *D* in which a hook at the end of the cord will engage when the hook is slipped along from *A* : this gives a ready means of throwing the indicator drum

in or out of gear. To make the mechanism fairly accurate the length of the pendulum rod AB should be considerably greater than that of the stroke. Another form of indicator gear is obtained by omitting BC, joining AB directly to the cross-head and setting the pin A on a sliding block in fixed guides which allow it to slide towards or away from the piston as the rod oscillates. This is geometrically a better form, but it requires careful construction to escape backlash at A.

Even when the cord which is to move the indicator drum is connected to the piston-rod in such a way as to copy its motion correctly, the motion of the drum itself may become incorrect because the length of the cord is not strictly constant. The varying pull causes varying amounts of extension, and when the cord is a long one the error which this involves may be serious. The tension in the cord varies from three causes:—(1) the varying resistance of the drum spring, (2) the varying acceleration of the drum, and (3) the friction of the drum (and guide pulleys, if there are any). Causes (1) and (2) may be arranged to counteract each other, but the friction of the drum necessarily tends to make the cord longer during the forward motion of the drum than during the backward motion. Hence it is important to see that the drum and pulleys turn readily with little friction, and still more important to make the cord short. Where there is a long distance between the indicator and the point from which motion is taken—as will generally be the case in large engines—cord should be used only at places where flexibility is required and stout wire should as far as possible be substituted. Even in comparatively small engines wire may be used with advantage[1]. Of all the errors to which indicator diagrams are liable perhaps none are so often neglected as those that come from the stretching of long driving cords[2].

130. Directions for taking Indicator Diagrams. In taking indicator diagrams the following practical hints may be found useful:—Before attaching the indicator to the engine, see

[1] See for example the device used by Prof. O. Reynolds on the experimental engine at Owens College, by which the length of the cord is reduced to a few inches (*Min. Proc. Inst. C. E.*, Vol. xc., 1889).

[2] For a discussion and experimental investigation of the errors of the indicator, see papers by Prof. O. Reynolds and Mr H. W. Brightmore (*Min. Proc. Inst. C. E.*, Vol. lxxxiii., 1896).

that the indicator is clean and in good order; that the piston moves very freely; that the joints of the lever and links are oiled with fine oil and are sufficiently slack to avoid friction, but not so slack as to allow the pencil to shake; that the pencil point is sharp, and that it is adjusted to press lightly upon the paper drum; and that the paper drum turns freely without shaking. The spindle on which the drum turns needs oil now and then.

Select a spring appropriate to the pressure within the cylinder and to the speed of the engine. With the Crosby indicator the diagram should not be more than 1¾ inches high; thus a 50 spring should not be used if the range of pressure to be indicated exceeds 87 lbs. per sq. in. When the engine runs fast it is necessary to use a still stiffer spring, to prevent the diagram from showing an inconvenient amount of oscillation. If large oscillations occur the process of smoothing the diagram by sketching a line midway between the crests and hollows is unsatisfactory, and a new diagram must be taken with a stiffer spring.

In putting a spring in and screwing the parts together, try whether there is any backlash or shake between the spring and the indicator's piston. If there is any it is to be taken up (in the Crosby instrument) by means of the set screw under the piston.

Screw the indicator cock to the pipe on the engine cylinder, and couple up the indicator, taking care to tighten up the coupling collar in such a position that it leaves the handle of the cock free to turn. See that the cord from the drum has a clear course to the oscillating lever, and that its mean position during the oscillation is about perpendicular to the lever. Adjust the length of the cord and the amount of its motion so that when the cord is in gear the drum turns backwards and forwards without coming up against a stop at either end of its travel. If it touches one stop or the other the cord is too long or too short: if it touches both stops the travel of the drum is too great and a point nearer the fulcrum of the oscillating lever must be taken for the attachment of the driving cord.

Do not keep the indicator drum moving except while diagrams are being taken. Stop the drum by disconnecting the cord from the oscillating lever before attempting to put a paper on the drum.

In putting on the paper see that it is taut and clear of wrinkles, and fold down the projecting edges so that they may not touch the lever which carries the marking pencil.

Turn on steam to the indicator for a minute or so before taking the diagram. Then press the pencil lightly on the paper, keeping it on long enough to complete a single diagram. Withdraw the pencil. Shut the cock leading to one end of the cylinder and open the cock from the other end (if pipes from both ends come to the same indicator). Touch the pencil to the paper again to take the other diagram. Withdraw it and shut the indicator cock. Touch the pencil again to the paper to draw the atmospheric line. Stop the drum by disconnecting the cord. Remove the paper and mark the diagrams to show which end of the cylinder each refers to. Note the scale number of the spring, and the speed of the engine, with the date and hour and any other particulars that may be wanted.

131. Calculation of the Indicated Horse-power. By measuring the mean height of the diagram between the top and bottom lines we find the *mean effective pressure,* which when multiplied by the area of the piston and the length of the stroke give the work done per stroke.

The mean height of the diagram is most accurately found by measuring the area of the diagram with a planimeter or otherwise and dividing that area by the length of the base, namely, the distance between lines drawn perpendicular to the atmospheric line and touching the diagram at its extremities. More usually the mean height is found by dividing the base into ten or twelve equal parts, drawing a perpendicular to the atmospheric line through the middle of each of these parts, measuring the lengths of these perpendiculars between the top and bottom lines and taking the mean of these lengths. The lengths of these perpendiculars are most conveniently measured by applying to each line in succession the edge of a scale graduated in inches and tenths of an inch: with a little practice it is easy to estimate to hundredths of an inch. The mean height in inches is multiplied by the scale number of the spring to find the mean effective pressure in lbs. per square inch. The mean effective pressure for the other side of the piston is found from the other diagram in the same way. Calling these mean effective pressures p_m and

p_m', in lbs. per square inch, and the net areas of the corresponding sides of the piston a and a' in square inches, and the length of the stroke l in feet, the work done by the steam per revolution is

$$l\,(p_m a + p_m' a')$$

in foot-pounds.

The work done per minute is

$$nl\,(p_m a + p_m' a'),$$

n being the number of revolutions per minute; and the indicated horse-power

$$\text{I.H.P.} = \frac{nl\,(p_m a + p_m' a')}{33000}.$$

In general a and a' are nearly equal. The mean of them may then be taken and multiplied by $p_m + p_m'$, as a substitute for the quantity within brackets. And it is convenient when many diagrams are to be worked out for one engine to express the quantity $\dfrac{l\,(a + a')}{2 \times 33000}$ as a single constant factor which has only to be multiplied by $n\,(p_m + p_m')$ to find the indicated power.

In place of the ordinary indicator an apparatus is occasionally used which integrates the two coordinates which it is the business of the indicator diagram to represent, and exhibits the power developed from stroke to stroke by the progressive movement of an index round a dial.

132. Examples of Indicator Diagrams. Fig. 54 shows a pair of indicator diagrams taken from a Corliss condensing engine, in which after a very early cut-off the whole expansion is performed

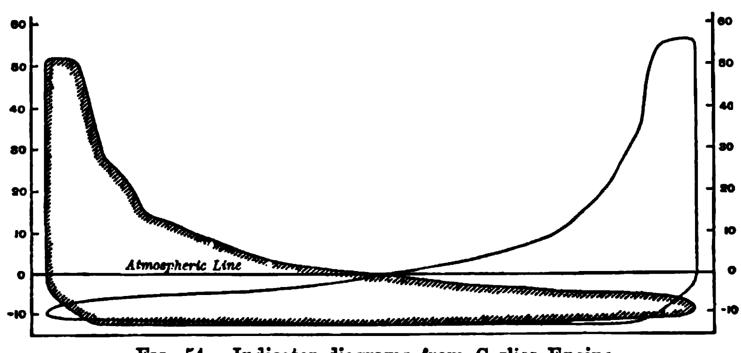


FIG. 54. Indicator diagrams from Corliss Engine.

in a single cylinder. In these and subsequent diagrams lines are added at either end which show the amounts of the respective clearances, and as base line the line of absolute vacuum is drawn, the distance of which below the atmospheric line is determined by reading the barometer. The numbers are pressures in lbs. per square inch above the atmosphere. Inspection of the diagram shows that the distribution of steam is very symmetrical as regards the two ends of the cylinder; also that the amount of compression might be increased with advantage. If an adiabatic curve be drawn through the point of cut-off (assuming a reasonable percentage of wetness) it will be found that the actual curve of expansion at first lies below the adiabatic curve but afterwards rises above it

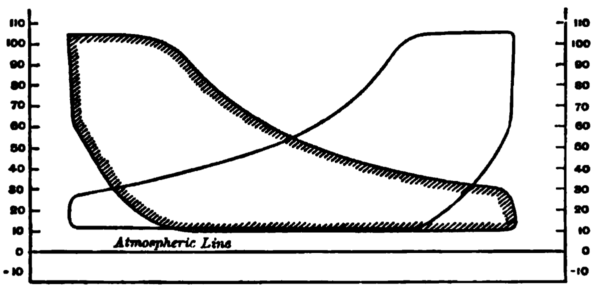


FIG. 55. Indicator diagrams from Compound Engine: High-pressure cylinder.

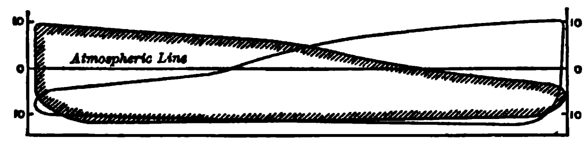


FIG. 56. Low-pressure cylinder.

in consequence of the re-evaporation of the condensed water (§ 109). Figs. 55 and 56 show a set of diagrams taken from a small compound engine using slide valves. Fig. 55 is the high-pressure pair of diagrams, and fig. 56 is the low-pressure pair. In the former the cut-off is a little sharper on one side than on the other, but the distribution is on the whole symmetrical and good. The points of release and compression are well marked, showing that there is a free exhaust. Other examples of compound diagrams will be given later.

Indicator diagrams are often taken for the purpose of testing the setting of the valves, although the circumstances may be such

that the engine is not doing external work. Fig. 57, for instance, is a pair of diagrams taken from a Corliss engine when first erected

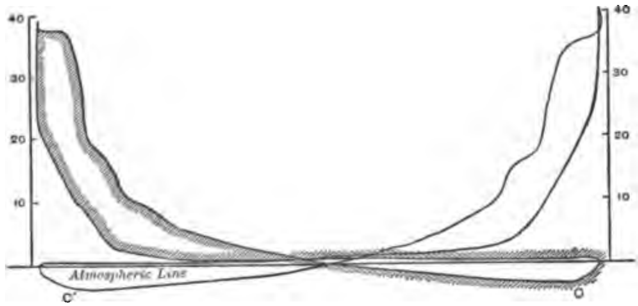


FIG. 57. Indicator diagrams taken to examine the action of the Valves.

by the makers without having the condenser in action and with no external load. The exhaust is into the atmosphere, and as expansion has made the pressure in the cylinder less than that of the atmosphere the pressure rises at release (C, C' in the figure). This produces a loop on the diagram representing negative work: the excess of the positive over the negative portion represents the net amount of work which is done by the steam in overcoming the friction of the engine.

133. Thermodynamic Tests. Measurement of the Supply of Steam by means of the Feed. When engine trials are to serve as tests of thermodynamic performance, either the heat supplied or the heat rejected has to be measured, for comparison with the work done. Measurements of the supply of heat are most usual. Sometimes, however, this method of testing may be impracticable and a test by means of the rejected heat may be easy. In any case a measurement of the rejected heat furnishes a valuable check on the accuracy of the other method, and the most satisfactory trials are made by measuring the heat supplied as well as the heat rejected; this allows a species of balance-sheet to be drawn up in which the heat given to the engine is more or less completely accounted for.

To determine the supply of heat the quantity of steam used by the engine is measured. Except when the engine has a surface-condenser, this has to be done by measuring the amount of feed-water that is required to keep the level of water in the boiler

constant during a prolonged run. A somewhat long run is necessary in a trial of this kind because the level of water in the boiler cannot be read very exactly and the whole consumption of feed-water must be so great that any error due to this cause will become negligible. With an ordinary Cornish or Lancashire boiler a run of six or eight hours may be desirable and even essential if an accurate result is to be got: on the other hand if the engine is getting its steam from a small tubular boiler working hard under forced draught, or from a water-tube boiler, the evaporation may be so rapid that a single hour or even less will suffice. Care should be taken to have all the conditions of the experiment as closely as possible the same at the end as at the beginning of the trial: if for instance the feed-pump is working at the beginning it should be working at the end at the same rate, and the pressure in the boiler should be the same. In these circumstances the quantity of water in the boiler, for a given reading in the gauge-glass, may be taken to be the same at the end as at the beginning of the run, and the quantity of feed-water that has been supplied in the interval is therefore equal to the quantity of steam (dry or wet) that has left the boiler. If there has been no leakage and no blowing off at the safety-valve or otherwise, this quantity of steam has been delivered to the engine.

To measure the feed-water a very convenient plan is to have two tanks, one a small tank (*A*, fig. 58) set above the other (*B*) so that it may drain into *B*. The weight of water contained by *A* when full must be accurately known, and it should be furnished with a gauge-glass *C* to let fractions of the whole contents be read. *B* must have a float or a point gauge *D* or other mark in it to indicate when the water reaches some one standard level. The feed-pump draws water from *B* by the pipe *E*; fresh water can be run into *A* at pleasure from the supply pipe *F*, and there is a stop-cock between *A* and

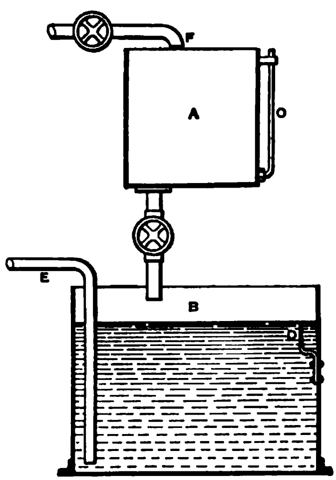


FIG. 58. Arrangement of tanks for measuring Feed-water.

B. At the beginning of the test let it be seen that the water in B is at the standard level and that the stop-cock between the tanks is shut. Supply water during the test by completely filling A as often as may be necessary, letting its contents drain completely into B each time and noting the hour and minute at which each fill of A is emptied into B. At the end of the trial, after filling A for the last time let just enough of its contents pass into B to bring the level of water in B up to the standard, and read on the gauge-glass of A the fraction which completes the whole supply. In a long run it is useful to check the work by dividing the whole period into two or more parts, in each of which the supply of feed-water is separately noted. The boiler pressure, the speed, and all other conditions of working must of course be kept as nearly uniform as may be throughout and should all be noted at regular intervals during the trial. It is useful to exhibit the log of the trial graphically by plotting all the observed quantities on section-paper with time as the base.

The engine should work for some time under the prescribed conditions as to speed, pressure, and load before the period of the test begins, in order that it may get thoroughly warmed up and that a uniform action may be established. During the trial indicator diagrams are taken from time to time and the times are noted. Where there is a mechanical counter the whole number of revolutions made during the period of trial is found by reading the counter at the beginning and at the end.

134. Measurement of the Supply of Steam by means of the Condensed Water. In engines which are fitted with a surface condenser the amount of steam passing through the engine in a given time is readily determined by weighing the condensed water discharged from the air-pump. An important advantage of this method is that a satisfactory trial of the engine can be made in much less time than is necessary when the steam used is to be determined from the feed-water. Provided the engine has been running long enough for the action to become uniform before the trial begins, the air-pump discharge need not be collected during more than ten or fifteen minutes, and thus a series of distinct trials under different conditions can be made in a single day.

135. Measurement of Jacket steam. If the engine has jackets the water condensed in them must be measured in addition

to the water discharged by the air-pump; and even when the whole supply of steam is inferred from the feed it may be desirable to determine separately the amount that is used in the jackets. This is done by draining them into a tank or tanks so that the condensed water may be weighed. The water must escape freely enough to prevent its accumulating in the jacket and yet not so freely as to let steam blow through. This is readily secured by means of one or other of the two devices shown in fig. 59. A gauge-glass is inserted in the jacket drain, or is fitted to the drain as in the left-hand figure, with a throttle valve below it. By adjusting this valve the escape of the condensed water can be regulated so that the surface of the water will show itself in the glass at a constant height; the water is then passing off just as fast as it is condensed. To prevent evaporation of the discharged water the continuation of the drain may pass in the form of a bend or worm through a tank of cold water so that the jacket water may be cooled before it reaches the vessel in which it is to be measured[1].

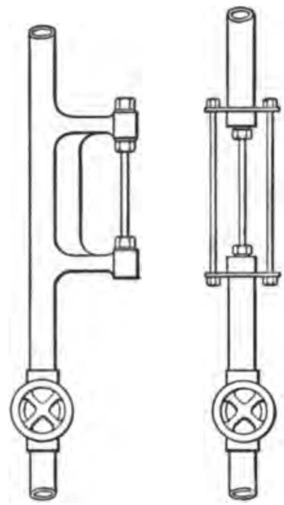


FIG. 59. Gauge on Jacket drain.

136. Comparison of Feed-water with Discharged Water. In many trials the quantity of steam used by the engine is measured by both of the means that have been described, namely, by finding on the one hand how much feed is supplied to the boiler, and on the other hand how much is discharged by the air-pump and the jacket drains. In most cases some discrepancy is observed: the feed-water may be as much as five per cent. more than the discharged water. This apparent loss of substance is due in part to water-vapour being discharged from the air-pump without being included in the measurement, but it is mainly due to leakage. Steam may escape at joints in small quantities showing little trace of its presence, and there is often some

[1] This device was shown to the author by Mr Bryan Donkin, who had used it in some of his engine tests. It is also used in the Barrus Calorimeter described below (§ 138).

leakage within the boiler, as for instance at the ends of the tubes into the firebox in a boiler of the locomotive or the marine type. In most cases the measurement of the water discharged by an engine gives a fairer test of its performance than is given by measuring the feed. Should a serious discrepancy between the two quantities be found its causes are of course to be searched for and remedied.

137. Estimation of Heat supplied. Measurement of Dryness of the Steam by the "Barrel" Calorimeter. Knowing the amounts of steam supplied to the cylinder and jackets we may go on to calculate the amount of heat which the working substance takes up. In the absence of information as to the proportion of water in the steam as supplied to the engine the assumption that the steam is dry is the only safe one, though this may do some injustice to the engine by over-estimating the supply of heat. When the dryness q is known the heat supplied per lb. is

$$qL + h - h_0,$$

h_0 being the heat already present in the feed-water. Direct measurement of q is difficult mainly because it is difficult to secure that the steam used in a test of dryness is of the same quality as that which is delivered to the engine. One method is to blow steam from the boiler into a barrel or other vessel containing water, allowing the steam to be condensed, and noting the amounts by which (1) the temperature and (2) the weight of the contents have become increased after a suitable time. The former shows how much heat has been given up in condensing the steam that is blown in; the latter shows what the quantity of that steam is. Let the temperature rise from t_1 to t_2 while the weight increases from W_1 to W_2. Then q is found from the equation

$$(W_2 - W_1)(qL + h - h_2) = W_1(h_2 - h_1),$$

where h_1 and h_2 refer to the temperatures t_1 and t_2, and h and L refer to the condition of the steam as supplied. This is subject to corrections (1) for loss of heat by radiation and (2) for the thermal capacity of the barrel itself. Accurate results are not easily got on account of the large error which is introduced by any inexactness in the measurement of the weight.

138. Barrus Calorimeter. A better form of calorimeter has been devised by Prof. Barrus, which also determines the wetness of

steam by measuring the heat given out during its condensation, but the condensed steam is not allowed to mix with the condensing water. The steam to be examined flows into a pipe which passes through a vessel of water and so forms a surface-condenser. A steady circulation of water is kept up in the vessel, cold water flowing in and passing off after having been warmed by the condensation of steam within the pipe. The temperatures t_1 and t_2 of the water at the inlet and outlet respectively are noted. The water formed by condensation in the pipe is weighed after allowing it to escape through a stop-cock furnished with a gauge-glass as in fig. 59 (§ 135) and its temperature t_3 is noted. The quantity of cooling water which passes through the vessel in a given time has also to be weighed. Before an observation is made the apparatus is kept running long enough to let the temperatures all take steady values. Then, if W be the quantity of cooling water which passes while the quantity w is condensed,

$$w(qL + h - h_3) = W(t_2 - t_1),$$

subject to a small correction for radiation as before, the amount of which can be determined by noting the rate at which the calorimeter cools when it stands full of water at temperatures intermediate between t_1 and t_2.

139. Measurement of the Wetness of Steam by means of Wire-drawing. Professor Peabody[1] describes a simple apparatus of his own design for measuring the proportion of water in steam, which acts by throttling the wet steam until it becomes dry or slightly superheated (see § 77). In this apparatus, which is commonly called a wire-drawing calorimeter, but to which the name calorimeter is scarcely appropriate, the steam passes through an adjustable throttle-valve A, fig. 60, into a chamber B lagged with non-conducting material, in which its temperature and its pressure are observed by thermometer C and gauge D. From this it escapes through another adjustable valve E to the atmosphere or to a condenser. The valves are adjusted until the steam in the chamber is seen to be slightly superheated, by comparing the observed temperature with the temperature which, in saturated steam, would correspond to the observed pressure. The amount

[1] *Thermodynamics of the Steam-Engine*, p. 237.

of superheating, and the drop in pressure which has caused it are noted. Let p_1 be the pressure in the chamber and t_1 the temperature which saturated steam at that pressure would have, and let t_1' be the actual temperature. Then

$$qL + h = L_1 + h_1 + \kappa (t_1' - t_1),$$

where κ is the mean specific heat of steam when superheated under constant pressure from saturation at t_1 to t'_1. The value of κ is uncertain when the amount of superheating is small (see § 90), and consequently in the use of this method it is highly important to keep $t_1' - t_1$ as small as possible. Unless this precaution is taken the method can lay claim to no accuracy. The usual practice of taking 0·48 as the value of κ probably leads to an under-estimate of the wetness, since κ may be expected in general to be larger than 0·48 in the first stages of superheating.

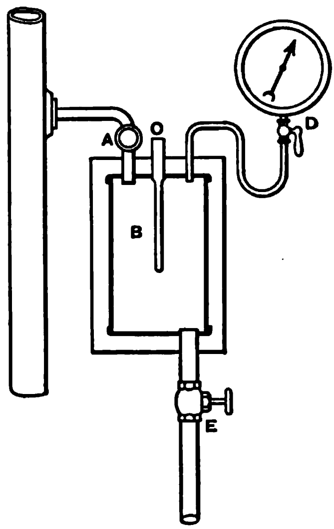


FIG. 60. "Throttling Calorimeter" (Peabody).

There is no need however to make t_1' greater than t_1 by more than a trifling amount—just enough to ensure that the steam in the chamber is perfectly dry. It is only when the steam is nearly dry to begin with that it can be superheated or even dried by throttling. Even when the steam is no more than dried by throttling the limit of wetness beyond which the apparatus cannot be used is not high. With steam at 100 lbs. pressure, for instance, only 4 per cent. of moisture can be removed by throttling if the pressure in the chamber is as low as that of the atmosphere: but if a condenser is available the pressure in the chamber may be reduced far enough to deal with about 6 per cent.

In a modified form of the apparatus by Prof. Barrus[1] the vessel B becomes a mere tube, separated from the steam-pipe by a diaphragm with a small aperture through which the steam is wire-drawn. A separator is added, between the throttling apparatus and the steam-pipe, to allow as much as possible of the

[1] *American Soc. Mech. Eng.*, 1890.

original moisture to be deposited before throttling takes place. With this addition it becomes possible to apply the method to steam that is originally very wet, for the separator leaves so little moisture still in the steam as to make throttling suffice to dry it completely. The water collected by the separator is to be added in reckoning the original wetness.

A porous plug forms a better means of throttling than the stop-valves and pin-hole orifices which have been used in instruments of this kind. The thermometer by which the temperature is taken after throttling should be placed as close as possible to the plug, for the steam quickly loses its superheat by conduction to the outside, and this cannot so well be done when a pin-hole orifice is used instead of a plug, since the kinetic energy of the stream through the orifice must be destroyed before it is allowed to come into contact with the thermometer. When all precautions are taken to secure that there shall be no losses of heat between the point of throttling and the thermometer, the method only serves, at the best, to show what was the wetness of the steam when it was on the point of entering the throttling plug. Whether its state then is the same as the average state of steam in the steam-pipe is another question. Unless special precautions be taken in the connexion of the whole apparatus to the steam-pipe, the sample of steam taken off for examination is liable to suffer some condensation before it reaches the plug and therefore to give an exaggerated impression of the wetness of the supply. On the other hand the steam supplied by a steam-pipe to an engine is liable to carry along with it a film of water on the inner surface of the pipe, and this wetness would not be represented in a sample taken off for the purpose of a test by means of a branch pipe opening into the interior of the steam-pipe in such a way as not to catch any of the water that is dragged along the surface by the current of steam. The consideration of these points will serve to show that little or no reliance can in general be placed on determinations of the general wetness of a steam supply by tests of a sample, whether the tests are made by the wire-drawing calorimeter or otherwise[1].

[1] See Prof. O. Reynolds on methods of determining the dryness of saturated steam. *Proc. Manchester Phil. Soc.*, 1896. For an account of various methods of determining the dryness of steam, see *Brit. Assoc. Report*, 1894, p. 392.

140. Measurement of Heat rejected by an Engine. The rejected heat is measured by observing the quantity of the condensing water and the amount by which its temperature rises as it passes through the condenser. With small engines the quantity may be found by direct weighing or measuring in a large tank, or by the use of a pair of measuring tanks arranged so that one fills while the other empties. But in general the quantity of condensing water is too great to be easily treated in this way, and it has rather to be gauged as a stream, by observing the *head* under which it flows through an orifice of known size, or over a weir. This gauging is generally done after the water leaves the condenser, in which case, if the condenser is of the injection type, the quantity that is measured is the sum of the cooling water and the condensed steam, and the amount of the cooling water alone can be inferred by deducting from the whole a measured or estimated allowance to represent the feed.

When the stream to be gauged is large an open weir with a rectangular or V-shaped notch will be found most convenient: but for small streams a submerged circular orifice has the advantage that the accuracy of the result is less affected by any small error that may be made in measuring the head. The stream to be gauged enters at *A* (fig. 61), a box containing baffle plates and perforated

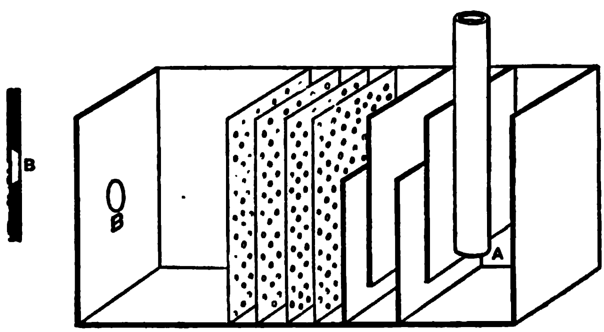


FIG. 61. Weir box with circular orifice.

diaphragms (sheets of perforated zinc or gauze will do well), which reduce it to stillness before it reaches the orifice *B*. This is a circular hole in a flat plate, and is bevelled to a sharp edge with the bevel outside. The head of water in the chamber close to the orifice is to be observed by means of a float and scale which are not shown

in the diagram. If h is the head in feet, measured from the surface to the centre of the hole, and s is the area of the hole in square feet, the discharge Q in cubic feet per second is given by the formula

$$Q = cs\sqrt{2gh},$$

where c is a "coefficient of discharge," the ordinary value of which for a circular hole in a large flat surface is 0·62, when the head is sufficient to bring the surface of the water to a considerable height above the hole[1].

A compact and simple weir-box may easily be made by using a tall rectangular sheet-iron tank, 10 or 12 inches square in horizontal section and 3 or 4 feet high. The pipe bringing in the water is brought down inside the tank, in a corner, and opens close to the bottom. The lower part of the tank is filled by sheets of gauze serving as baffle plates. The water rises steadily above these, and escapes by one or more sharp-edged circular orifices cut in the side of the tank about a foot from the top.

In trials of marine engines where the weir-box method of measuring is impracticable, the feed-water may be measured with considerable accuracy by interposing a water-meter between the feed-pump and the boiler.

141. Example of an Engine Trial. To illustrate the reduction of the observations and the comparison of the heat supplied with the heat rejected and the work done we may take the data of a test by Mr Mair-Rumley, from one of the papers which were alluded to in last chapter. The engine under trial was a compound beam-engine with steam-jackets and with a jet-condenser. The cylinders were 21 and 36 inches in diameter, and the stroke of each piston was $5\frac{1}{2}$ feet. The feed-water was measured during a period of 6 hours and the air-pump discharge

[1] When an open rectangular notch with sharp edges in a vertical plate is used for a weir, h is to be measured from the bottom of the notch to the free level of the surface, at a distance far enough back to give practically still water; then

$$Q = 3\cdot33\,(b - 0\cdot2h)\,h^{\frac{3}{2}},$$

where b is the breadth of the notch.

With a triangular notch cut so that the breadth is twice the depth $Q = 2\cdot54h^{\frac{5}{2}}$, where h is the depth of the bottom of the notch below the still-water surface level. For the justification of these formulas reference must be made to books on hydraulics or to papers by James Thomson, *Rep. Brit. Assoc.* 1858, 1861, and 1876, p. 243.

was gauged by means of a weir. The following are the data of the trial:—

Pressure in boiler, 76 lbs. per sq. in., absolute (for which $L = 898$ and $h = 278$).
Duration of trial, 6 hours.
Revolutions, 8632, or 24·0 per min.
Indicated horse-power, 127·4.
Feed-water, 12032 lbs.
Air-pump discharge, 1226 lbs. per min.
Water drained from jackets, 1605 lbs.
Dryness of steam as supplied, 0·96.
Temperature of feed, $t_0 = 59°$ Fah.
" " injection $t_1 = 50°$ Fah.
" " air-pump discharge, $t_2 = 73·4°$ Fah.

These give the following results :—

Total feed per revolution $= 1·394$ lbs.
Jacket feed per revolution $= 0·186$ lbs.
Cylinder feed per revolution $= 1·208$ lbs.
Injection water per revolution $= \frac{1226}{24} - 1·208 = 49·9$ lbs.

Heat taken in by the working substance per revolution

$$= 1·394\,(qL + h - h_0)$$
$$= 1·394\,(0·96 \times 898 + 278 - 27) = 1·394 \times 1113$$
$$= 1551 \text{ thermal units.}$$

Heat converted into work per revolution

$$= \frac{127·4 \times 42·42^{1}}{24} = 225 \text{ thermal units.}$$

The whole heat rejected per revolution should therefore be 1326 thermal units.

That part of the working substance which is cylinder feed rejects heat first and chiefly to the injection water, and secondly by becoming itself cooled from t_2 the temperature of the air-pump discharge to t_0 the temperature at which it returns to the boiler. The heat it rejects per revolution in these two ways is therefore

$$49·9\,(t_2 - t_1) + 1·208\,(t_2 - t_0),$$

or $$49·9 \times 23·4 + 1·208 \times 14·4 = 1184 \text{ thermal units.}$$

[1] 42·42 is the thermal equivalent of 1 horse-power acting for 1 minute, namely, $\frac{33000}{778}$ thermal units.

That part of the substance which is jacket steam rejects heat by becoming cooled from the temperature at which it is condensed in the jacket to the temperature at which it is returned to the boiler. In the present case the jackets drained into the hot-well and the temperature therefore fell to 59°, the temperature of the feed. The heat rejected in this way per revolution was

$$0{\cdot}186\,(h-h_0)=0{\cdot}186\,(278-27)=47 \text{ thermal units.}$$

Adding these we have 1231 units of rejected heat. A balance of 95 units remains to be accounted for. It is made up partly of heat carried away by the air and vapour of the air-pump discharge, partly of losses through radiation from the engine and pipes, and partly of heat lost in whatever steam escapes by leakage. In the example cited the loss by radiation was estimated to amount to 45 units[1]; allowing for this the discrepancy between the two sides of the account is reduced to 50 units or only about 3 per cent. of the whole supply.

The consumption of steam per indicated horse-power-hour, calculated from the whole amount of the feed is

$$\frac{12032}{127{\cdot}4\times 6} \text{ or } 15{\cdot}7 \text{ lbs.}$$

This makes the indicated work done per lb. of steam equivalent to

$$\frac{33000\times 60}{778\times 15{\cdot}7} \text{ or } 162 \text{ thermal units.}$$

In considering the efficiency of a cycle as a whole we should in strictness deduct from this the net amount of work which has to be expended in returning the condensed steam from the condenser to the boiler, or say $0{\cdot}017\times 76\times 144$ foot-lbs. per lb. As this is the equivalent of only 0·24 thermal units per lb. the correction is unimportant. Since the heat taken in per lb. is 1113 units the efficiency of the cycle is 0·145[2].

[1] The loss by radiation is approximately estimated by letting the engine stand still with the jackets and steam-chest full of steam and noting the amount that is condensed in a given time.

[2] For further illustrations of engine trials and the reduction of results reference should be made to the excellent examples contained in several of Mr M. Longridge's *Reports* as Engineer of the Engine, Boiler, and Employers' Liability Association from 1880.

142. Wetness of the steam during expansion. In § 109, Chapter V., it was explained how to find the proportion of water present in the cylinder at any stage of the expansion and to represent the results of this calculation graphically by means of a "saturation curve" upon the indicator diagram—namely, a curve which represents the volume which the steam in the cylinder should fill at any pressure if it were dry throughout. To draw this curve for either side of the piston we should in strictness know how the whole amount of the cylinder feed is shared by the two ends of the cylinder—a matter which the test does not determine. But in general the action in the two ends is so nearly symmetrical that results which are practically correct may be obtained by combining the indicator diagrams for the two into a mean diagram, taking for clearance the mean of the two actual clearances, and taking half the cylinder feed per revolution as the quantity of steam that enters the cylinder per stroke. The diagram shown in fig. 45, § 109, is in fact a combination diagram drawn in this way. To determine the wetness of the steam during expansion is an important part of an engine test, and the results cannot be better exhibited, so far as this particular is concerned, than by showing the saturation curve in its relation to the actual curve of pressure and volume. In dealing with compound engines a saturation curve may be drawn separately for each cylinder, or the diagrams for the several cylinders may be combined into one by means of a device which will be described in the next chapter.

This process of estimating the water present during expansion by comparing the saturation volume with the volume actually filled by the working substance depends on the assumption that the whole quantity of substance does not change from the time that cut-off is complete until release begins. Any leakage of steam, in or out, through the valve or past the piston will invalidate the calculation.

143. Transfer of Heat between the Steam and the Metal. Hirn's Analysis. Having determined what proportion of the working substance is steam and what is water throughout the expansion we may go on to calculate how much heat is taken from or given to the walls of the cylinder and piston during any stage of its action. This analysis of the transfers of heat,

introduced by Hirn and developed by his pupils and followers, has been pursued at great length in some engine tests[1]. Only a very short account of it need be given here.

Let m and m' represent the quantities of dry steam and water respectively present in the working mixture either during expansion or during compression. We may use I to represent the internal energy of the whole mixture. Its value at any stage is

$$(m+m')h+m\rho,$$

ρ having the meaning which was assigned to it in § 60. Taking any two points in the curve of expansion (or in the curve of compression) let the corresponding two values of I be calculated, say I_1 and I_2. Between these two points a quantity of work is done by the steam (or upon it, if the compression stage is being considered) which is measured by $\int PdV$, where P and V represent the actual pressure and volume of the mixture and the integral is taken between limits corresponding to the two assumed points. Call this quantity of work W_{12}. If it happens that $W_{12}=I_1-I_2$ the process is adiabatic: no heat in that case has been taken from or given to the cylinder walls by the working steam between the two points. More generally there will be a difference between the work done and the change of internal energy, which difference measures the quantity of heat that is transferred to or from the walls. Thus if we distinguish the four events of admission, cut-off, release and compression by the suffixes a, b, c and d respectively, the heat taken up from the cylinder walls during expansion is

$$Q_{bc}=W_{bc}-(I_b-I_c).$$

Similarly the quantity

$$Q_{da}=W_{da}-(I_d-I_a),$$

in which the values of I relate to the substance shut up in the clearance space, measures the heat that is taken up during compression. W_{da} is negative. This calculation can of course be applied to any stage of either process, and thus by applying it to a series of short stages a curve showing the inflow or outflow of heat can be drawn from point to point of the stroke.

During admission the quantity of the mixture is undergoing change. The mixture that is shut up in the clearance at the end

[1] See Dwelshauvers-Dery, "*Étude Calorimétrique de la Machine à Vapeur.*" Also Mr Mair-Rumley's papers already cited.

of the back stroke before admission takes place has a certain internal energy I_a. The steam that enters brings with it an additional quantity of energy, H_0, which may be calculated provided the dryness of the entering steam is known. H_0 is made up of the internal energy of the steam supplied, together with the work spent on it as it enters the cylinder, or $\frac{P_0V_0}{J}$. The work done by the steam, up to cut-off, W_{ab} is determined from the diagram. Then the transfer of heat during admission is

$$Q_{ab} = W_{ab} - (I_a - I_b + H_0),$$

a quantity which is generally negative in actual cases since heat is given to the cylinder walls in this part of the action.

In attempting to apply the same method of calculation to determine the heat taken up from the metal during exhaust (Q_{cd}) we are met by the difficulty that the state as regards wetness in which the mixture leaves the cylinder is not known. The value of Q_{cd} may however be estimated indirectly as follows. Let Q_{ab}, Q_{bc} and Q_{da} represent, as before, the transfer of heat from metal to steam during admission, expansion and compression respectively, let Q_r represent the loss by radiation and Q_j the additional supply of heat which is furnished by condensation of steam in the jacket, all reckoned per stroke. Then if the engine is working uniformly the gains and losses of heat on the part of the metal must balance, and hence $Q_{cd} = Q_j - Q_r - Q_{ab} - Q_{bc} - Q_{da}$.

This heat Q_{cd} which is taken up from the metal during the exhaust is called in the writings of Hirn and his pupils "le refroidissement au condenseur," and is sometimes spoken of as being in a particular sense the measure of the wasteful action of the cylinder walls. It should however be borne in mind that the transfer of heat between metal and steam does some mischief even when the steam is dry at the end of expansion, in which case practically no heat is taken up during exhaust. That part of the heat abstracted from the steam during admission which is restored before release does not appear in Q_{cd}, nevertheless it reduces the efficiency because it is taken from the working substance at a high temperature and restored at a lower. And this action goes on even when the steam is so dry at release that Q_{cd} is sensibly zero.

As an alternative to the method used by Hirn, the entropy-temperature diagram may be applied to the purpose of tracing the

heat stored and restored during expansion, in a manner which has been sufficiently indicated in Chapter V. (§ 110).

144. Tests of mechanical efficiency. Measurement of Brake Horse-power. In tests of mechanical efficiency the engine (unless it be used for pumping) is commonly set to work against some form of friction brake arranged to serve as an absorption dynamometer. For engines of small or moderately small power no form is so simple or so easy of application as a rope or band brake of the type shown in fig. 62. Two, three, or more parallel turns of rope with a few wood blocks to hold them apart (the number of ropes depending on the quantity of power that is to be absorbed) are made to clasp the fly-wheel in the manner sketched; the slack end is attached to a spring balance and the other end is loaded with weights, either directly, or through a lever if the amount of load is inconveniently great. A little grease applied to the surface of the metal makes the brake work quietly and steadily. The wheel may be kept cool by water; a wheel with internal flanges on the rim, forming an internal channel to which cold water may be supplied, is convenient. The resistance is adjusted by varying the amount of the weight T_1. A platform or stop fixed a little way below this weight allows the brake to be applied and removed by pulling or slacking away the rope by which the spring balance is suspended; by pulling the rope which is shown at the top of the figure the weights are lifted off their platform and the brake comes into action. When the brake is in action the pull T_2 indicated by the spring balance is noted from time to time. The effective resistance is $T_1 - T_2$, and the work done against the brake per revolution is $2\pi r (T_1 - T_2)$ where r is the radius measured from the axis of rotation to the middle of the rope's thickness. Hence the brake horse-power

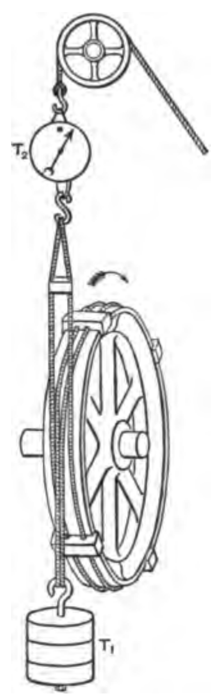


FIG. 62.

$$\text{B. H.-P.} = \frac{2\pi n r (T_1 - T_2)}{33{,}000}.$$

The mechanical efficiency is the fraction $\frac{\text{B. H.-P.}}{\text{I. H.-P.}}$; or, without reducing to horse-power, it is the ratio of the work done on the brake per revolution to the work done by the steam per revolution, namely,

$$\frac{2\pi r(T_1 - T_2)}{l(p_m a + p_m' a')},$$

in the notation of § 125.

A flexible band such as may be made by using a few strips of cotton listing has the advantage as compared with rope of working smoothly and silently without any lubrication and is to be preferred to rope for small engines. When only two or three horse-power have to be measured a single strip of listing will be found to make an excellent brake.

In dealing with large powers the most effective and accurate absorption dynamometer is one in which the work of the engine is spent in churning water by turning a species of turbine wheel in a casing through which water is continuously passed. The casing is held from turning by applying weights to a lever arm, and this measures the moment exerted by the engine-shaft, on which the turbine wheel is fixed. Prof. Reynolds has designed and used in his experiments a very perfect brake of this kind, a full description of which will be found in the *Philosophical Transactions of the Royal Society* for 1897. He has successfully applied this brake not only in engine trials but also to measure the mechanical equivalent of heat by observing, along with the work done, the quantity of water which in passing through the brake had its temperature raised from 32° F. to 212° F.

145. Trials of an engine under various amounts of load. Although some engines are required to work always under the same or nearly the same conditions as to load, more commonly the load is liable to variation, and it may be as important to examine the performance under light loads as to make trials at full power. At electric light stations, for example, much of the work is done with a comparatively light load on the engine and the efficiency under these conditions is a matter of the greatest moment. To be complete a trial should include a series of tests made at various grades of output from full power down to the extreme when the engine merely drives itself without doing external work.

The papers by Mr Willans which were referred to in Chapter IV. contain many examples of trials which are complete in this sense. Such results may be represented graphically by drawing a curve in which the ordinates are the number of lbs. of steam consumed per horse-power-hour, with the rate of output in horse-power for abscissæ.

Two curves of this kind are shown in fig. 63, relating to two series of tests by Willans of one of his compound high-speed

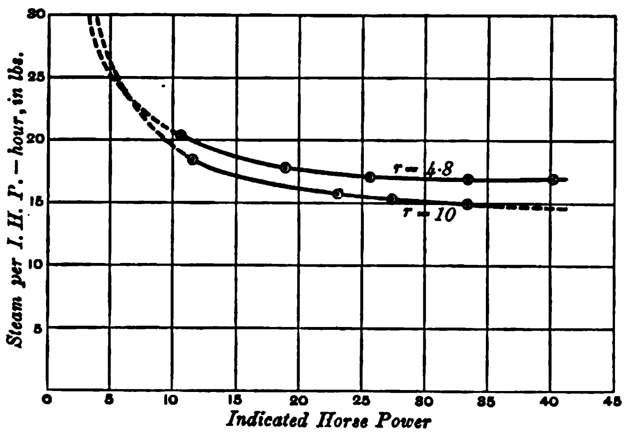


FIG. 63.

single-acting engines, using a condenser[1]. In one set of trials the ratio of expansion was 4·8 and the points through which the curve is drawn were determined by testing the consumption under various values of the initial steam pressure, ranging from 135 lbs. per sq. inch (absolute) down to 43 lbs. The other curve refers to a similar series of trials in which the ratio of expansion was 10.

Another useful way of showing the performance at all powers is to plot the whole quantity of steam consumed per hour in relation to the horse-power. Curves of this kind were first used by Willans: examples of them are given in fig. 64 relating to the same two sets of trials as fig. 63. In each set of trials the adjustment of the power was accomplished by varying the initial pressure of the steam, the cut-off remaining constant throughout the set, and the speed of the engine also remaining constant.

[1] *Min. Proc. Inst. C. E.*, Vol. cxiv. 1893.

Under these conditions Willans found that the curve of total steam consumption in relation to power (fig. 64) was sensibly a straight line. With variable cut-off and constant pressure, on the other hand, the Willans' line is curved, having a steeper gradient at high powers than at low powers.

The same types of diagram are useful in representing the consumption of steam in relation to brake horse-power, pump horse-power, electrical horse-power, etc. They exhibit clearly under what condition the maximum of efficiency will be reached,

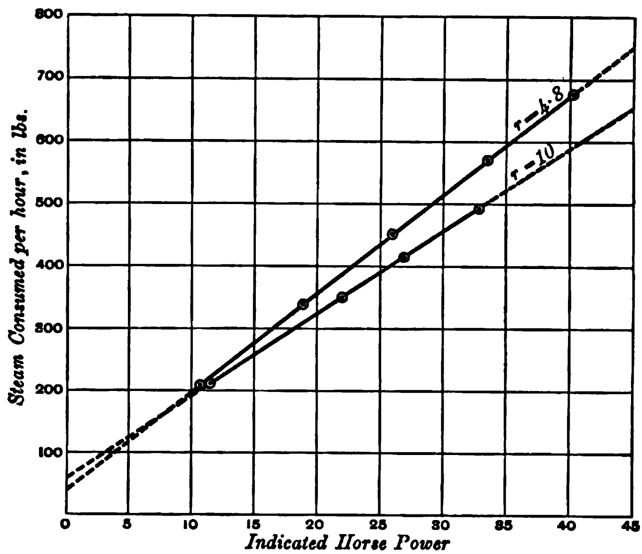


Fig. 64.

and also what the performance will be under the less favourable conditions that may have to be submitted to in practice.

When the Willans' line (fig. 64) is a straight line the whole consumption of steam at any load may be regarded as made up of two parts—the constant unproductive consumption that takes place without doing work in the cylinder and a further consumption that is simply proportional to the indicated power. The whole consumption is equal to

$$a(i + b),$$

where i is the number of horse-power and a is the ratio at which steam is taken per horse-power after the unproductive supply ab has been furnished.

The same remark holds good in relation to a Willans' line drawn with the external or brake horse-power as the base; *ab* then represents the steam that is used in making the engine drive itself; and *b* is what may be called the "idle work," a quantity which is somewhat greater than the indicated work done in overcoming engine friction.

A comparison of the Willans' lines relating to indicated and brake power respectively serves to show how far the work spent on engine friction remains constant at high powers and at low. In general it may be expected that this quantity will be greater at high powers since the forces at the joints of the mechanism become on the whole increased.

An example of the Willans' line drawn for trials of a steam turbine at various grades of output, but at constant speed, will be found in Chapter XII. The line there is not far from straight, but has a slight curvature of the same kind as is found in ordinary engines when the output is varied by changing the position of cut-off.

CHAPTER VII.

COMPOUND EXPANSION.

146. Woolf Engines. When the expansion of steam is begun in one cylinder and continued in another, the steam may either be made to pass directly from one cylinder to the next, or it may pass from the first cylinder into an intermediate chamber, called a "receiver" from which the second cylinder draws its supply. An advantage of the latter plan is that it does not require the reception of steam by the second cylinder to be simultaneous with the rejection of steam by the first. This allows the cranks to be set at any angle, it also allows the distribution of the expansion between the two cylinders to be readily adjusted. Chiefly for this reason compound engines are now rarely used with immediate transfer of steam from one cylinder to the other.

The original form of compound engine invented by Hornblower and revived by Woolf had no receiver. Steam passed directly from the high to the low pressure cylinder, entering one as fast as it was exhausted from the other. This arrangement is possible only when the high and low pressure pistons begin and end their strokes together, that is to say, when their movements either coincide in phase or differ by half a revolution. Engines of the "tandem" type satisfy this condition—engines, namely, of which the high and low pressure cylinders are in one line, with one piston-rod common to both pistons. Engines whose high and low pressure cylinders are placed side by side, and act either on the same crank or on cranks set at 180° apart, may also discharge steam directly from one to the other cylinder; the same remark applies to beam engines with high and low pressure cylinders standing side by side. By a convenient usage which is now pretty general the name "Woolf engine" is restricted to those compound engines

which discharge steam directly from the high to the low pressure cylinder without the use of an intermediate receiver.

147. Receiver engine. An intermediate receiver becomes necessary when the phases of the pistons in a compound engine do not agree. With two cranks at right angles, for example, a portion of the discharge from the high-pressure cylinder occurs at a time when the low-pressure cylinder cannot properly receive steam. The receiver is in some cases an independent vessel connected to the cylinders by pipes; very often, however, a sufficient amount of receiver volume is afforded by the valve casings and the steam-pipe which connects the cylinders. The receiver, when it is a distinct vessel, is frequently jacketed.

The use of a receiver is by no means restricted to engines in which the "Woolf" system of compound working is impracticable. On the contrary, it is frequently applied with advantage to beam and tandem compound engines. Communication need not then be maintained between the high and low pressure cylinders during the whole of the stroke: in such cases admission to the low-pressure cylinder is stopped before the stroke is completed; the steam already admitted is allowed to expand independently; and the remainder of the discharge from the high-pressure cylinder is compressed into the intermediate receiver. Each cylinder has then a definite point of cut-off, and by varying the cut-off in the low-pressure cylinder the distribution of work between the two cylinders may be adjusted at will. In general it is desirable to make both cylinders of a compound engine contribute equal or nearly equal quantities of work. If they act on separate cranks this has the effect of giving the same value to the mean twisting moment for both cranks.

Another adjustment which is sometimes aimed at is to make the range of temperature equal in both. In general, when the division of work is equal, the parts into which the whole temperature range is divided are nearly equal also.

148. Drop in the Receiver. Compound diagrams. Wherever a receiver is used, care must be taken that there is no large amount of unresisted expansion into it; in other words, the pressure in the receiver should not be greatly above that in the high-pressure cylinder at the moment of release. Any drop in the steam pressure between the high-pressure cylinder and the

receiver will show itself in an indicator diagram by a sudden fall at the end of the high-pressure expansion. This "drop" is, from the thermodynamic point of view, irreversible, and therefore wasteful. Practically some small amount of drop is desirable for the same reasons which make a rather incomplete expansion preferable to complete expansion in the working of a single cylinder. The drop can be reduced to any desired extent, or wholly avoided, as we shall presently see, by selecting a proper point of cut-off in the low-pressure cylinder. When there is no "drop" the expansion that occurs in a compound engine has precisely the same effect in doing work as the same amount of expansion would have in a simple engine, provided the law of expansion be the same in both and the waste of energy which occurs by the friction of ports and passages in the transfer of steam from one to the other cylinder be negligible. The work done in either case depends merely on the relation of pressure to volume throughout the process: and so long as that relation is unchanged it is a matter of indifference whether the expansion be performed in one vessel or in more than one. It has, however, been explained in Chapter V. that in general a compound engine has a thermodynamic advantage over a simple engine using the same pressure and the same expansion, inasmuch as it reduces the exchange of heat between the working substance and the cylinder walls and so makes the process of expansion more nearly adiabatic. The compound engine has also a mechanical advantage which is referred to in § 153, below.

The ultimate ratio of expansion in any compound engine is the ratio of the volume of the low-pressure cylinder to the volume of steam present in the high-pressure cylinder at the point of cut-off.

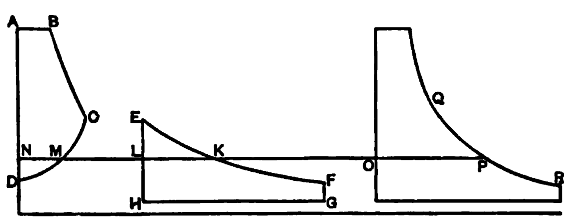


FIG. 65. Compound Diagrams: Woolf type.

Fig. 65 illustrates the combined action of the two cylinders in a hypothetical compound engine of the Woolf type, in which for simplicity

the effect of clearance is neglected and also the loss of pressure which the steam undergoes in transfer from one to the other cylinder. *ABCD* is the indicator diagram of the high-pressure cylinder. The exhaust line *CD* shows a falling pressure in consequence of the increase of volume which the steam is then undergoing through the advance of the low-pressure piston. *EFGH* is the diagram of the low-pressure cylinder, and is drawn alongside of the other for convenience in the construction which follows. It has no point of cut-off; its admission line is the continuous curve of expansion *EF*, at each point of which the pressure is the same as at the corresponding point in the high-pressure exhaust line *CD*. At any point *K*, the actual volume of the steam is $KL+MN$. By drawing *OP* equal to $KL+MN$, so that *OP* represents the whole volume, and repeating the same construction at other points of the diagram, we may set out the curve *QPR*, the upper part of which is identical with *BC*, and so complete a single diagram which exhibits the equivalent expansion in a single cylinder. The area of the figure so drawn is equal to the sum of the areas of the high-pressure and low-pressure diagrams.

In a tandem compound engine of the receiver type the diagrams resemble those shown in fig. 66. During *CD* (which

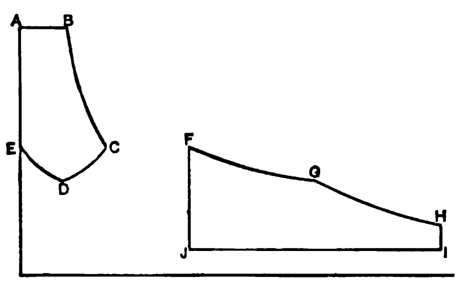


FIG. 66. Compound Diagrams: Receiver type.

corresponds to *FG*) expansion is taking place into the large or low-pressure cylinder. *D* and *G* mark the point of cut-off in the large cylinder, after which *GH* shows the independent expansion of the steam now shut within the large cylinder, and *DE* shows the compression of steam by continued discharge from the small cylinder into the receiver. At the end of the stroke the receiver pressure is *OE*, and this must be the same as the pressure at *C*, if

there is to be no 'drop.' In the diagram sketched it is assumed that there is none. The case of 'drop' would be illustrated if we were to cut off the corner at C by a vertical line drawn from some earlier point in BC to meet the curve CD; this would of course also imply a shortened high-pressure stroke. Diagrams of a similar kind may be sketched without difficulty for the case of a receiver engine with any assigned phase-relation between the pistons.

It may be noticed in passing that an intermediate receiver has the thermodynamic advantage that it reduces the range of temperature in the high-pressure cylinder, and so helps to prevent initial condensation of the steam. This will be made obvious by a comparison of fig. 65 and fig. 66. The lowest temperature reached in the high-pressure cylinder is that corresponding to the pressure at D, and is materially higher in fig. 66 than in fig. 65.

149. Adjustment of the division of work between the cylinders, and of the drop. Graphic method. By making the cut-off take place earlier in the large cylinder we increase the mean pressure in the receiver; the work done in the small cylinder is consequently diminished. The work done in the large cylinder is correspondingly increased, for the total work (depending as it does almost wholly on the initial pressure and the total ratio of expansion) is unaffected or scarcely affected by the change. Hence we have the apparently anomalous result that a shorter admission to the low-pressure cylinder causes it to do a larger share of the whole work.

Further, the same adjustment—namely, hastening the cut-off in the low-pressure cylinder—serves, in case there is 'drop,' to remove it. By selecting suitable values of the ratio of cylinder volumes to one another and to the volume of the receiver, and also by choosing a proper point for the low-pressure cut-off, it is possible to secure absence of drop along with equality in the division of the work between the two cylinders.

To determine beforehand that point of cut-off in the low-pressure cylinder which will prevent drop when the ratio of cylinder and receiver volumes is assigned is a problem most easily solved, or approximately solved, by a graphic process. The process consists in drawing the curve of pressure during admission to the low-pressure cylinder until it meets the curve of expansion which

is common to both cylinders[1]. In fig. 67 (where for the sake of simplicity the effects of clearance are neglected) *AB* represents the admission line and *BC* the expansion line in the small cylinder. Release occurs at *C*, and from *C* to *D* steam is being taken by the large cylinder. *D* corresponds to the cut-off in the

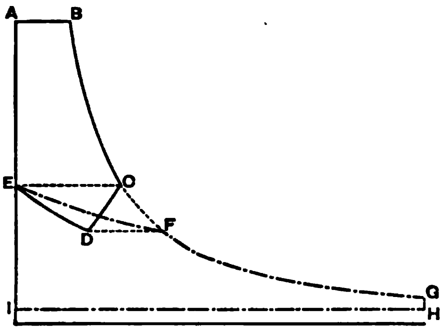


Fig. 67.

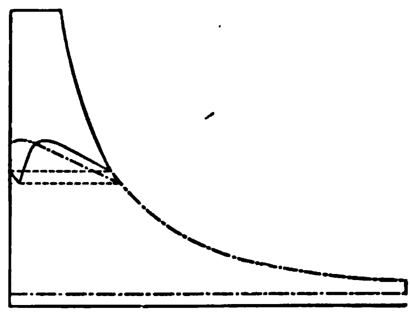


Fig. 68.

Figs. 67 and 68.—Determination of the point of Cut-off in the low-pressure cylinder of a compound engine.

large cylinder, which is the point to be found. From *D* to *E* steam is being compressed into the receiver. To avoid drop the receiver pressure at *E* is to be the same as the pressure at *C*. *E* is therefore known, and may be employed as the starting-point in drawing a curve *EF* which is the admission line of the low-pressure diagram *EFGHI*. This line is drawn by considering at each point in the low-pressure piston's stroke what is then the whole volume of the steam. The place at which *EF* intersects the continuous expansion curve *BCG* determines the proper point of cut-off. The sketch (fig. 67) refers to the case of a tandem receiver engine; but the process may also be applied to an engine with any assumed phase-relation between the cranks. Fig. 68 shows a pair of theoretical indicator diagrams determined in the same way for an engine with cranks at right angles, the low-pressure crank leading. In these examples the volume of the receiver has been taken equal to the volume of the high-pressure cylinder. With a larger receiver the variations of pressure during the back stroke of the high-pressure piston would be less conspicuous. In using the graphic method any form may be assigned

[1] See a paper by Prof. R. H. Smith, "On the Cut-off in the Large Cylinder of Compound Engines," *The Engineer*, November 27, 1885.

to the curve of expansion. Generally this curve may be treated without serious inaccuracy as a common hyperbola, in which the pressure varies inversely as the volume. The construction may obviously be applied to triple and quadruple expansion engines. For an accurate solution it would be necessary to take the effect of clearance into account and also to allow for some loss of pressure in the passage from one vessel to another. The figures given here omit these complications, and treat the expansion as hyperbolic.

150. Algebraic Method. When this simple relation between pressure and volume is assumed, it is not difficult to find algebraically the low-pressure cut-off which will give no drop, with assigned ratios of cylinder and receiver volumes. Taking the simplest case—that of a tandem engine, or of an engine with parallel cylinders whose pistons move together or in opposition—we may proceed thus. Since the point of cut-off to be determined depends on volume ratios we may for brevity treat the volume of the small cylinder as unity. Let R be the volume-ratio of the receiver to the small cylinder, and L the volume-ratio of the large to the small cylinder. Let x be the required fraction of the stroke at which cut-off is to occur in the large cylinder; and let p be the pressure at release from the small cylinder. If there is to be no drop, p is also the pressure in the receiver at the beginning of admission to the large cylinder. During that admission the volume changes from $1 + R$ to $1 - x + R + xL$, and the pressure at cut-off is therefore $\frac{p(1+R)}{1-x+R+xL}$. The steam that remains is now compressed into the receiver, from volume $1 - x + R$ to volume R. Its pressure therefore rises to

$$\frac{p(1+R)}{1-x+R+xL} \cdot \frac{(1-x+R)}{R},$$

and this, by assumption, is to be equal to p. We therefore have

$$(1+R)(1-x+R) = R(1-x+R+xL),$$

whence
$$x = \frac{R+1}{RL+1}.$$

Thus, with $R = 1$ and $L = 3$, cut-off should occur in the large cylinder at half-stroke (which is the case illustrated by the diagram of fig. 66); with a greater cylinder ratio the cut-off

in the large cylinder should be earlier, as it is, for instance, in fig. 67.

A similar calculation[1] for a compound engine whose cranks are at right angles, and in which cut-off occurs in the large cylinder before half-stroke, shows that the condition of no drop is secured when

$$2R(xL - 1) = 1 - 2\sqrt{x(1-x)}.$$

In some compound engines a pair of high-pressure cylinders discharge into a common receiver; in some a pair of low-pressure cylinders are fed from a receiver which takes steam from one high-pressure cylinder, or in some instances from two. With these arrangements the pressure in the receiver may be kept much more nearly constant than is possible with the ordinary two-cylinder type. Occasionally compound engines work without any mechanical connexion between the cranks, and the pressure within the receiver then depends not only on the adjustment of the points of cut-off but also on the relative frequency of stroke of the pistons.

151. Ratio of Cylinder Volumes. The size of the low-pressure cylinder in a compound engine is fixed by reference to the power the engine is intended to develope, the speed, the given boiler pressure, and the total ratio of expansion. But the size of the high-pressure cylinder remains a matter of choice when all these things are settled. Say that the total ratio of expansion is to be r; we may choose any ratio L less than r for the volume-ratio of the large to the small cylinder. It will then be necessary to make the cut-off in the small cylinder happen at a fraction of the stroke equal to $\frac{L}{r}$ in order that the final volume of the steam, when it fills the whole of the large cylinder, may be r times its initial volume up to the point of cut-off in the small cylinder. Thus an earlier or later adjustment of the cut-off in the high-pressure cylinder will allow the whole ratio of expansion to take whatever value may be wanted, no matter what be the ratio of the cylinder volumes.

[1] Examples of calculations dealing with particular arrangements of two and three cylinder compound engines will be found in an Appendix to Mr R. Sennett's *Treatise on the Marine Steam-Engine.*

Again, as we have seen above, by varying the cut-off in the large cylinder we can adjust matters so that equal amounts of work are done in both cylinders, irrespective of their sizes.

But it is only when a suitable ratio of volumes has been selected that this adjustment to equalise the work will also secure a reasonable absence of 'drop'—or that an adjustment of the low-pressure cut-off to avoid drop will not too seriously disturb the balance of work.

This consideration serves to fix in a general way the proper proportion of the volumes. No hard and fast rule is followed; an exact balance in the work is not essential, and a complete absence of drop is not even desirable. The same practical considerations which make it undesirable in a simple engine to have complete expansion apply in regard to compound engines: unless there is some little drop the last part of the stroke is ineffective. It should also be remembered that drop in a compound engine is not quite so wasteful as it looks: the unresisted expansion into the receiver serves to dry the steam and in extreme cases even to superheat it.

Another consideration enters into the question. In some engines, especially marine engines, it is a point of importance to avoid having an early cut-off in any of the cylinders, partly to avoid unnecessarily severe stresses in the mechanism and partly to allow the valves to be of the simplest kind. This may lead to the existence of more drop than would otherwise be permissible. In practice the choice of volume ratios is to some extent a compromise between conditions that are more or less incompatible, and, as might be expected, a good deal of variety is found.

In a two-cylinder compound condensing engine, for instance, using steam of 80 or 90 lbs. pressure the large cylinder may have from three to four times the volume of the small cylinder. The steam in this case should expand about 12 times; if a ratio of 3 to 1 be chosen the conditions of equal work and very little drop will be secured by putting the cut-off at something like one-fourth of the stroke in the high-pressure cylinder and at about one-sixth of the stroke in the low-pressure cylinder. An example will be found in the indicator diagrams given below in fig. 70. On the other hand, if the high-pressure cylinder have only one-fourth of the volume of the other, a later cut-off will serve. The suitable ratio of volumes

depends on the boiler pressure; thus if it is $3\frac{1}{2}$ with 70 lbs. it may be as much as $4\frac{1}{2}$ with 100 lbs.

In triple-expansion engines, where the boiler pressure is rarely less than 150 nor more than 180 lbs., the third cylinder has usually, in marine practice, from 6 to 7 times the capacity of the first cylinder, and the second cylinder has from $2\frac{1}{4}$ to $2\frac{3}{4}$ times that of the first. In land engines of this type, where an earlier cut-off may be resorted to without inconvenience, the first cylinder may be rather larger: its capacity ranges from about one-fifth to one-sixth that of the low-pressure cylinder.

152. Advantage of Compound Expansion in the economical use of High-Pressure Steam. The thermodynamic advantage of compound expansion has been pointed out in § 119. It allows high-pressure steam to be used without the excessive waste which would occur if a high grade of expansion were attempted in a single cylinder. So long as the boiler pressure does not much exceed 100 lbs. this advantage is sufficiently secured by dividing the expansion into two stages: accordingly the ordinary compound engine or two-stage expansion engine is used with pressures up to 100 lbs. but seldom with higher pressures. Beyond this triple expansion becomes in general advisable if the full benefit of the higher pressure is to be secured. But when the expansion is divided into three stages it becomes advantageous to use a pressure considerably higher than the limit we have just named: thus with triple engines a pressure of 160 to 170 lbs. is usual. Intermediate pressures, of say 120 or 130 lbs., are not often found: they are too high to suit the two-cylinder compound engine and too low to let triple expansion give its best effects. Quadruple expansion has little if any advantage when the pressure is under 200 lbs.; up to this pressure and perhaps beyond it the thermodynamic benefit of a fourth stage is scarcely sufficient to justify the mechanical complication it involves. With the types of boilers that are ordinarily used this limit of pressure is rarely exceeded and the quadruple expansion engine is not common. In naval practice the use of water-tube boilers has in some cases raised the pressure of steam to 250 and even to 300 lbs., but in general three stages of expansion are preferred to a larger number of stages.

153. Mechanical advantage of Compound Expansion. Uniformity of Effort in a Compound Engine. A simple engine using high-pressure steam with an early cut-off has the drawback, from the mechanical point of view, that the thrust of the steam on the piston during the early part of the stroke is very great in comparison with the mean thrust. The initial pressure of the steam acts on the full area of a piston whose size is determined by reference to the mean pressure. The piston and connecting rod, the framing and other parts of the machine must be made strong enough for this relatively great initial thrust, also there is much wear and tear at joints, and for steady motion a large fly-wheel becomes necessary.

The compound engine avoids the extreme thrust and pull which would have to be borne by the piston-rod of a single-cylinder engine working at the same power with the same initial pressure and the same ratio of expansion. If all the expansion took place in the low-pressure cylinder, the piston at the beginning of the stroke would be exposed to a thrust greater even than the sum of the thrusts on the two pistons of a compound engine of equal power. Thus in the tandem engine of fig. 65 the greatest sum of the thrusts will be found to amount to less than two-thirds of the thrust which the large piston would be subjected to if the engine were simple. The mean thrust throughout the stroke is of course not affected by compounding; only the range of variation in the thrust is reduced. The effort on the crank-pin is consequently made more uniform, the strength of the parts may be reduced, and the friction and wear at joints lessened. Thus even in a tandem compound engine there is mechanically some advantage, and the benefit of compounding in this respect is obviously much greater when the cylinders are placed side by side, instead of tandem, and work on cranks at right angles. As a set-off to its advantage in giving a more uniform effort, the compound engine has the draw-back of requiring more working parts than a simple engine with one cylinder. But in many instances—as in marine engines—two cranks and two cylinders are in any case almost indispensable, to give a tolerably uniform effort and to get over the dead-points without the aid of a heavy fly-wheel; and the comparison should then be made between a pair of simple cylinders and a pair of compounded cylinders. Another point in favour of the compound engine is that, although the whole ratio of expansion is great,

there need not be a very early cut-off in either cylinder; hence the common slide-valve, which is unsuited to give an early cut-off, may be used in place of a more complex arrangement. The mechanical advantages of compound working were recognized sooner than its thermodynamic economy, and did much to bring it into favour before, indeed, the practice had grown up of using steam high enough in pressure to make compounding very distinctly economical.

Again, apart from its improved economy the mechanical merits of the triple engine have contributed much to bring it quickly to the position it now holds in marine practice. The advantage of three cranks over two in giving uniform effort and comparatively little friction and wear is conspicuous, and a triple engine with its three cranks set at 120° from each other is now the standard marine type.

154. Examples of Indicator Diagrams from Compound Engines. Fig. 69 shows a pair of diagrams from the two cylin-

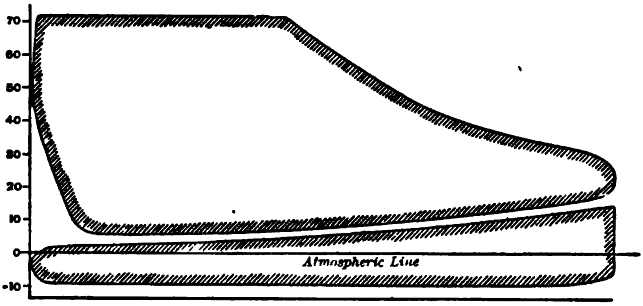


FIG. 69. Indicator diagrams of a Woolf Engine.

ders of a Woolf engine, in which the steam passes as directly as possible from the small to the large cylinder. Both pistons have the same length of stroke. The diagrams are drawn to the same scale of stroke and therefore to different scales of volume, and the low-pressure diagram is turned round so that it may fit into the space below the high-pressure diagram. There is some drop at the high-pressure release, and further the friction of the passages causes the admission line of the large cylinder to lie slightly lower than the exhaust line of the small cylinder. The transfer of steam goes on throughout nearly the whole of the back stroke until compression begins in the small cylinder. The steam then present

in the large cylinder continues expanding for the small part of the stroke that is left until the point of release is reached.

An example of compound diagrams for an engine of the receiver type has already been given in figs. 55 and 56, Chap. VI.

The receiver in that engine was unusually large, which accounts for the nearly level line drawn during the back stroke of the small piston. Another example is given in fig. 70, which shows the diagrams of a tandem receiver engine with cylinders 30 and 52 inches in diameter and 6 ft. stroke (volume ratio 1 to 3), taking steam at an initial pressure of 80 lbs. above the atmosphere. With this proportion of volumes and with the somewhat early cut-off shown by the diagrams there is a complete absence of any objectionable drop and a nearly equal division of work between the cylinders. Expansion valves (see Chap. VIII.) were used to

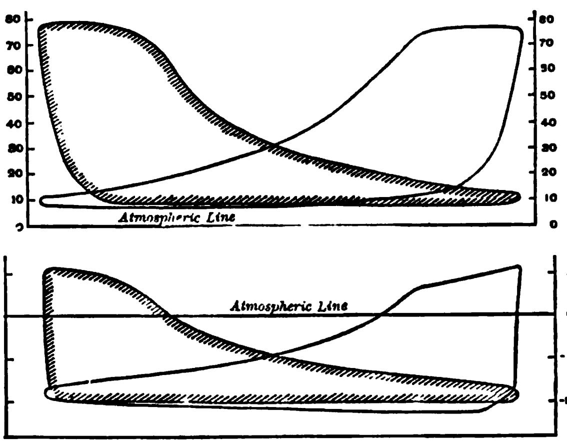


FIG. 70.

produce this early cut-off. The exhaust line of the small cylinder dips in the middle, as in fig. 67, but much less, for here the receiver is more capacious. When the cranks are set at right angles this line rises towards the middle, as fig. 68 indicates.

Fig. 71 shows a set of triple expansion diagrams, from trials (by a Committee of the Institution of Mechanical Engineers) of the steamship "Iona." The cylinder diameters were 21·9 in., 34 in. and 57 in., giving a volume ratio of 1 : 2·4 : 6·8, and the stroke was 39 in. The engines made 61 revolutions per minute and developed 208 I. H. P. in the first cylinder, 217 in the second and

220 in the third, with a consumption of 13·35 lbs. of steam per I. H. P.-hour. A simple slide-valve was used on each cylinder[1].

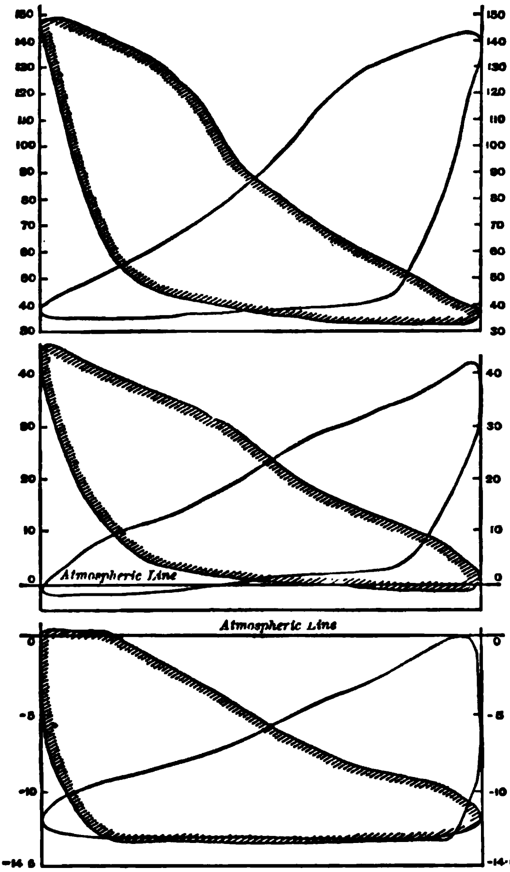


FIG. 71. Indicator diagrams of Triple-expansion Engine.

155. Combination of the Indicator Diagrams in Compound Expansion. The indicator diagrams of a compound engine may be combined in such a way that the pressures and volumes in the several cylinders are displayed in proper relation to one another, by the use of a single scale of pressures and a single scale of volumes. Some care, however, is necessary in the interpretation

[1] Report of the Research Committee on Marine Engine Trials, *Proc. Inst. Mech. Eng.*, April, 1891.

of such combined diagrams, and the construction to be adopted will depend on the use that is aimed at.

A common practice is to set out each diagram from the line of no volume through a distance which represents the clearance in the corresponding cylinder. This is illustrated in fig. 72, which has been drawn to exhibit in combination the diagrams already shown in figs. 55 and 56, § 132. Each of the two diagrams in fig. 72 is a mean for the two sides of the piston, and the distance of each from the line OY is the mean clearance in the corresponding cylinder. Diagrams drawn in this way are not without their uses, but it must be remembered that the

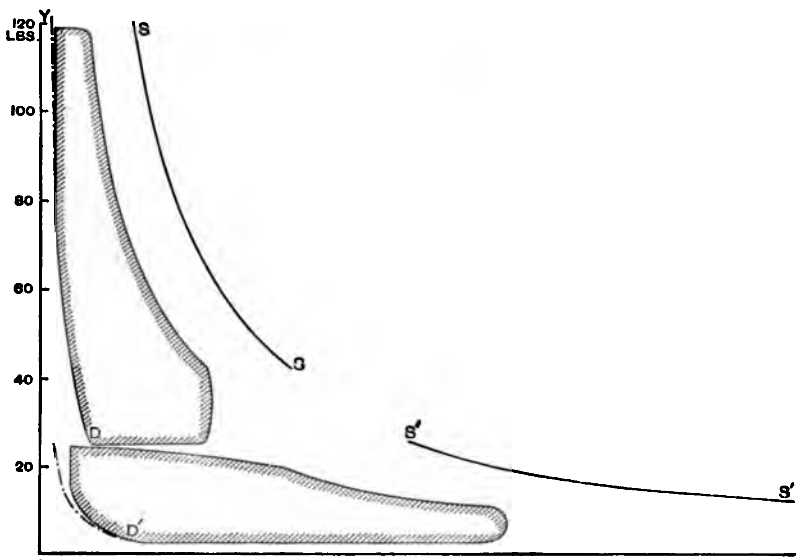


FIG. 72.

amount of substance which is taking part in the expansion is different in the two parts of the combination, and consequently a single adiabatic curve or a single saturation curve cannot properly be drawn to apply to both. The line SS is the saturation curve for the first stage of expansion, and the line $S'S'$ for the second stage. In this example the cylinder feed per single stroke was 0·0498 lbs., and the cushion steam was 0·0074 lbs. in the small cylinder and 0·0022 lbs. in the large cylinder. The saturation curve SS is accordingly drawn for 0·0572 lbs. and $S'S'$ for 0·052 lbs.

The amount of the substance present in the cylinder is in

general different in the successive stages because of differences in the amount of cushion steam in the several cylinders: the cylinder feed is the same throughout. If therefore we modify the diagram in such a way as to eliminate the cushion steam, leaving the cylinder feed only, we may draw a single saturation curve which will serve for all the expansion.

This is done in fig. 73, which represents the same pair of diagrams, transformed by the following device. From points D, D' (fig. 72) taken at the places where compression has begun and the exhaust is complete, saturation curves are drawn for the cushion steam in the respective cylinders. These curves are indicated by

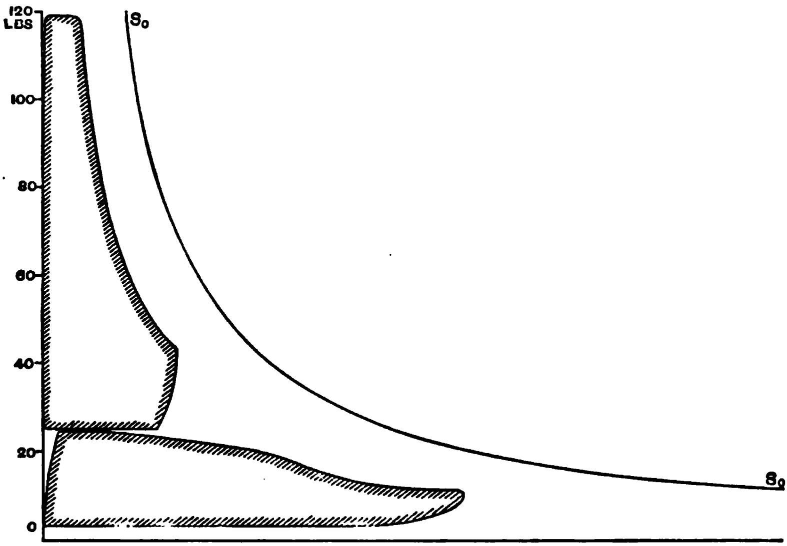


FIG. 73.

broken lines in the figure: the one that relates to the small cylinder is scarcely distinguishable from the compression curve of the indicator diagram. The diagrams are then redrawn as in fig. 68, using horizontal distances from these curves as abscissæ. This is equivalent to subtracting from the actual volumes throughout the diagram a quantity which represents the volume the cushion steam would occupy if it were saturated at all pressures. The result is that the area of the diagram remains unaltered: its area is still a true measure of the work. But a single saturation curve S_0S_0 may now be drawn—namely, for a quantity of steam equal to the cylinder feed—which will apply equally to both (or all) stages of

the compound expansion. The horizontal distance at any pressure between the expansion curve in fig. 73 and the saturation curve S_0S_0 is the same as the horizontal distance at that pressure between the expansion curve in fig. 72 and its corresponding saturation curve. It still represents the volume which has disappeared by condensation or what is often called the 'missing quantity.' The chief advantage of this construction is that it makes a single saturation curve possible, and so allows the changes in the amount of water present to be readily exhibited as the steam passes through the whole course of its expansion.

This will be apparent from figs. 74 and 75, which are copied from Professor Osborne Reynolds' account of triple engine trials[1], to which reference was made in Chapter V. Here the cushion steam has been eliminated in the manner just described and a single saturation curve has been drawn for the cylinder feed. The horizontal width of shaded space between the actual expansion curves and this line measures the water present at any stage in the expansion. Fig. 74 refers to a test made without steam in

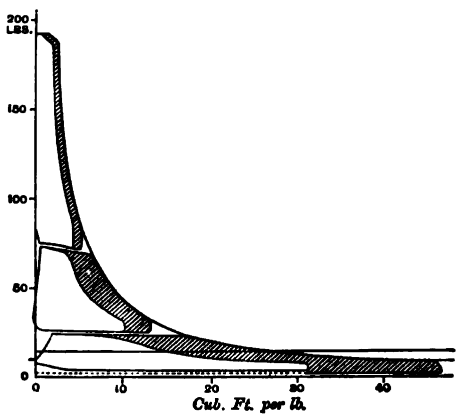


FIG. 74.

the steam-jackets, and fig. 75 to a test when all the jackets were supplied with steam at the full boiler pressure of 190 lbs. The drying influence of the jacket is conspicuous: in fig. 75 there is scarcely any condensation in the third cylinder.

These diagrams relate to an engine built for experimental use in which the three pistons could move independently, at different

[1] *Min. Proc. Inst. C. E.*, Vol. XC.

speeds, and the speeds were in fact different. Hence to prepare the diagrams for combination a further device was employed: the

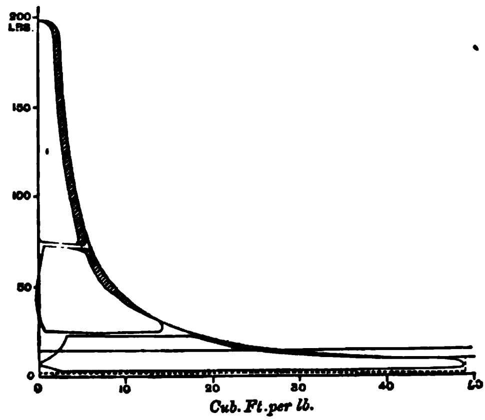


FIG. 75.

common scale of length of the diagrams was chosen so that the volumes represented in each are reckoned per lb. of cylinder feed. The scale of volume is accordingly divided in the figures to show cubic feet per lb. of water passing through the engine. This is a

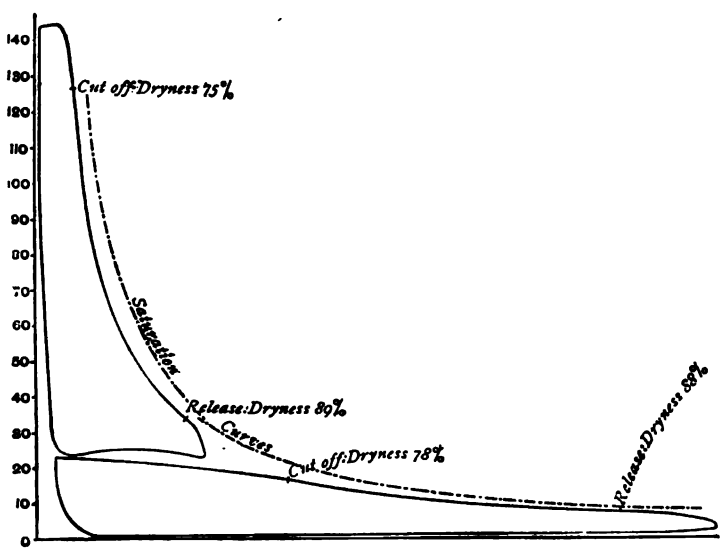


FIG. 76.

method of graduation which might be followed with advantage even in ordinary cases, where it is not rendered necessary by the pistons having independent speeds, for it facilitates comparison between various trials.

An additional example of compounded indicator diagrams is given in fig. 76, which represents in a combined form the diagrams reproduced in § 124, figs. 47 and 48, relating to a trial by Mr Longridge of a two-cylinder compound engine where slow running and efficient jacketing made the amount of condensation in the cylinders considerably less than in the example of fig. 72. The diagrams are set out separately as in that figure, with the clearance appropriate to each, and the two corresponding saturation curves are drawn. Here the cylinders were 17 and 34 inches in diameter and the stroke was 5 ft. The mean clearance volume was 0·26 cubic feet in the high-pressure cylinder, and 0·84 cubic feet in the low.

CHAPTER VIII.

VALVES AND VALVE-GEARS.

156. The Slide-Valve. In early steam-engines the distribution of steam was effected by means of conical lift-valves, rising and falling on conical seats, and worked by tappets from a rod which hung from the beam. The slide-valve, the invention of which is credited to Murdoch, an assistant of Watt, came into general use with the introduction of locomotives, and is now employed, in one or other of many forms, in the great majority of engines.

The common or locomotive slide-valve is illustrated in fig. 77, which shows a sectional side and end elevation and a plan. The seat, or surface on which the valve slides, is a plane surface formed

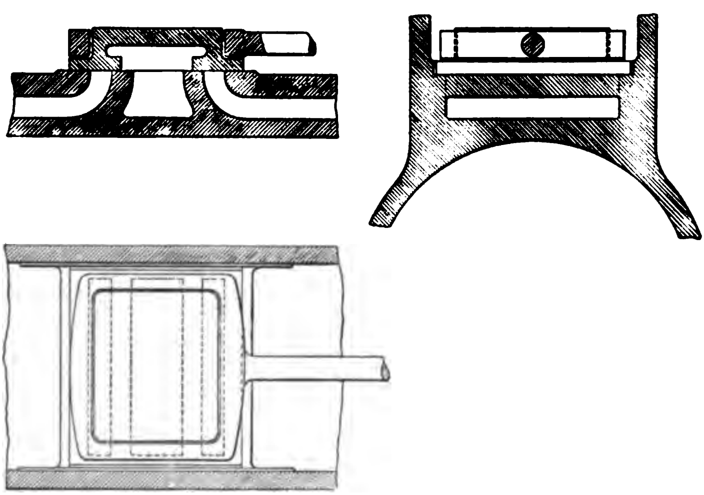


FIG. 77. Common Slide-Valve.

on or fixed to one side of the cylinder, with three ports or openings, which extend across the greater part of the cylinder's width. The ports are shown in the plan by dotted lines. The central opening is the exhaust-port through which the steam escapes; the others, or steam-ports, which are narrower, lead to the two ends of the cylinder respectively. The valve is a box-shaped cover which slides upon the seat, and the whole is enclosed in a chamber called the valve-chest, to which steam from the boiler is admitted. The valve is pulled backwards and forwards across the ports by means of a valve-rod which passes out of the valve-chest through a steam-tight stuffing-box. The valve is attached to the valve-rod, not rigidly but in such a way that, while it has no longitudinal freedom to slide along the rod, it is free to take a close bearing on the seat, under the pressure exerted by the steam on its back. In its middle position the valve covers both steam-ports completely, but when it is moved a sufficient distance to either side of the middle position, it allows fresh steam to enter one end of the cylinder from the valve-chest, and allows the steam which has done its work to escape from the other end of the cylinder through the cavity of the valve into the exhaust-port. The valve-rod is generally moved by an eccentric on the engine-shaft, which is mechanically equivalent to a crank whose radius is equal to the eccentricity, or distance of the centre of the shaft from the centre of the eccentric disc or sheave. The sheave is encircled by a strap to which the eccentric-rod is fixed, and the rod is connected by a pin-joint to the valve-rod outside of the valve-chest. The eccentric-rod is generally so long that the motion of the valve is sensibly the same as that which it would receive were the rod infinitely long. Thus if a circle (fig. 78) be drawn to represent the path of the eccentric-centre during a revolution of the engine, and a perpendicular PM be drawn from any point P on a diameter AB, the distance CM is the displacement of the valve from its middle position at the time when the eccentric-centre is at P. AB is the whole travel of the valve.

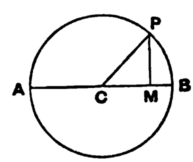


FIG. 78.

157. Lap, Lead, and Angular Advance. If the valve were formed so that when in its middle position it did not overlap

the steam-ports (fig. 79), any movement to the right or the left would admit steam, and the admission would continue until the valve had returned to its middle position, or, in other words, for

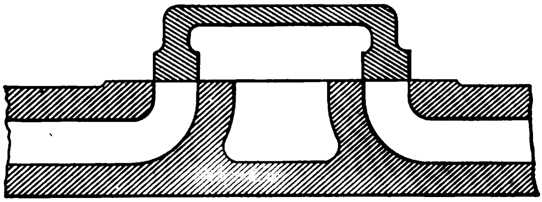


FIG. 79. Slide-Valve without Lap.

half a revolution of the engine. Such a valve would not serve for expansive working; it would admit steam to one end of the cylinder during all the stroke, and at the same time would exhaust steam from the other end during all the stroke. As regards the relative position of the crank and eccentric it would have to be set so that its middle position was coincident in point of time with the extreme position of the piston; in other words, the eccentric radius would have to make a right angle with the crank.

To make expansive working possible the valve must be able to keep the cylinder ports closed during some part of the stroke. For this purpose it must have what is called *lap*, that is to say its edges must project beyond the ports as in fig. 80, where *e* is the *outside lap* and *i* is the *inside lap*. Admission of steam to either

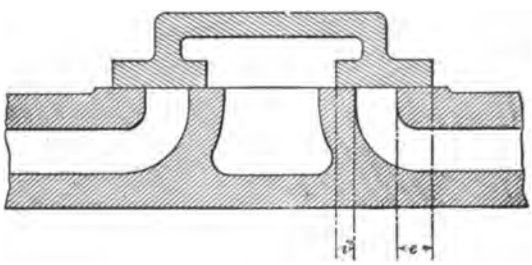


FIG. 80. Slide-Valve with Lap.

end of the cylinder now begins only when the displacement of the valve from its middle position is equal to the outside lap, and continues only until the valve returns to the same distance from its middle position. Further, exhaust begins only when the valve has moved past the middle position by a distance equal to the inside lap and continues until the valve has again returned to

this distance from its middle position. Thus let a circle (fig. 81) be drawn to represent the path of the eccentric-centre, on a

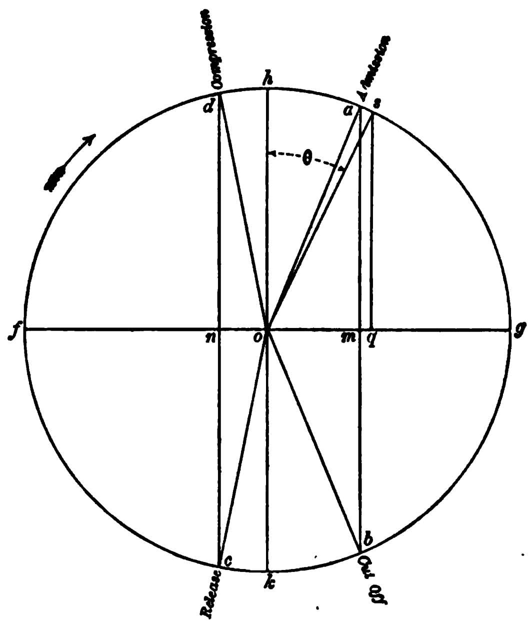


FIG. 81.

diameter *fg* which is the whole travel of the valve, let *om* be set off equal to the outside lap *e* and *on* to the inside lap *i*, and let perpendiculars *amb* and *cnd* be drawn at these distances from the centre. The points *a*, *b*, *c* and *d* then mark the positions of the eccentric-centre at which the four events of admission, cut-off, release and compression respectively occur for one end of the cylinder. As to the other end the four events are determined in the same way by setting off the corresponding outside lap to the left of *o* and the inside lap to the right of *o*. The laps may or may not be equal for the two ends of the cylinder. For the sake of clearness we may for the present confine our attention to one of the two. Of the whole revolution the part from *a* to *b* is the arc of admission; in other words, the port is open to steam while the shaft turns through an angle equal to *aob*. Similarly *bc* is the arc of expansion, *cd* that of exhaust and *da* that of compression.

The relation of these events to the piston's position is still undefined. If the eccentric were set in advance of the crank by

an angle equal to *foa*, the valve would be just beginning to open as the piston stroke begins. It is, however, desirable, in order to allow the steam free entry, that the valve should be already some way open when the piston stroke begins, and hence the eccentric is set at a rather greater angular distance in advance of the crank. Thus if the angular position of the eccentric is *os* while the crank is at the dead-point (on the line *of*) the valve is already open by the distance *mq*, which is called the *lead*. The angle θ by which the whole angle between the crank and the eccentric exceeds a right angle is called the *angular advance*, this being the angle by which the eccentric is set in advance of the position it would hold if the primitive arrangement without lap were adopted. The lap e, the lead l, the angular advance θ, and the half-travel or throw of the eccentric r are connected by the equation

$$e + l = r \sin \theta.$$

An effect of lead is to cause *preadmission*, that is to say, the lead allows steam to enter before the back stroke is quite completed, and this increases the mechanical effect of the compression in "cushioning" the piston during the reversal of its motion.

The greatest amount by which the valve is ever open during the admission of steam is the distance *mg*. The width of the steam port is made at least equal to this distance, and is often greater in order that the wider opening *nf* which occurs during exhaust may be taken advantage of.

158. Graphic method of examining the distribution of steam given by a slide-valve. Let the circle *APB* (fig. 82) represent the path of the crank-pin about the centre *O*, the stroke

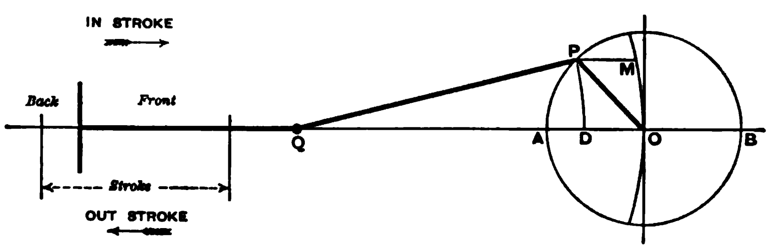


Fig. 82.

being *AB*. When the crank is at any point *P* the position of the piston may be found by projecting the point *P* on *AB* by drawing

a circular arc *PD* with the length of the connecting-rod *PQ* as radius and the cross-head *Q* as centre. Then *DO* represents the displacement of the piston from its middle position, and *AD* and *DB* represent its distance from the two ends of the stroke. Another construction equivalent to this is to draw through *O* the arc *OM* with the length of the connecting-rod as radius, and draw *PM* parallel to *AB*. *PM*, being equal to *DO*, measures the displacement of the piston from its position at mid-stroke. In speaking of the two ends of the cylinder we shall distinguish the one nearer the crank as the front end and the other as the back end. The stroke towards the crank may be called the in-stroke and the other the out-stroke, as marked in fig. 82.

To find the position of the piston at each of the four events we have to make a construction which is equivalent to transferring

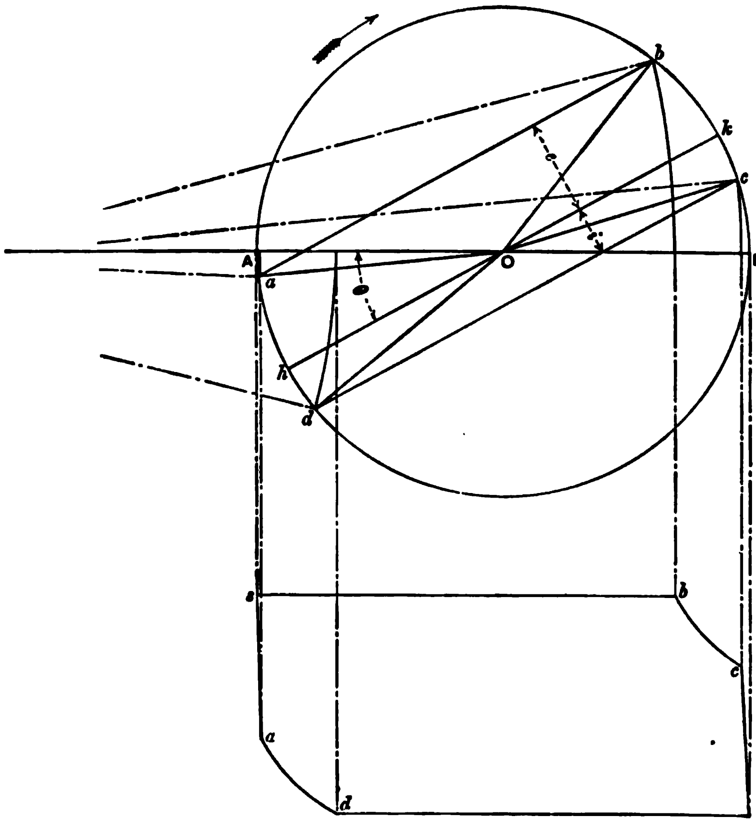


Fig. 83.

from fig. 81 the four positions of the crank which correspond to the positions a, b, c, d of the eccentric. This is most readily done by drawing a single circle (fig. 83) to represent the motion of the crank-pin on one scale and the motion of the eccentric-centre on another scale. Taking the diameter AB to represent the piston stroke, draw another diameter hk to represent the line hk of fig. 82 turned back through an angle of $90° + \theta$, so that the angle AOh (fig. 83) is equal to θ. Draw ab and cd parallel to this line at distances from it equal to the outside and inside laps respectively. The effect is that each of the points a, b, c and d is turned back, in fig. 83, through an angle equal to $90° + \theta$ as compared with its position in fig. 82. Consequently these points in fig. 83 show the positions which the crank has at the four events. And the corresponding positions of the piston may be found by projecting the points a, b, c, d on AB by means of circular arcs. This is shown in fig. 83, and the indicator diagram is also sketched by reference to the positions projected on AB.

The following is an equivalent and rather more convenient construction. Let a circle (fig. 84) be drawn as before to

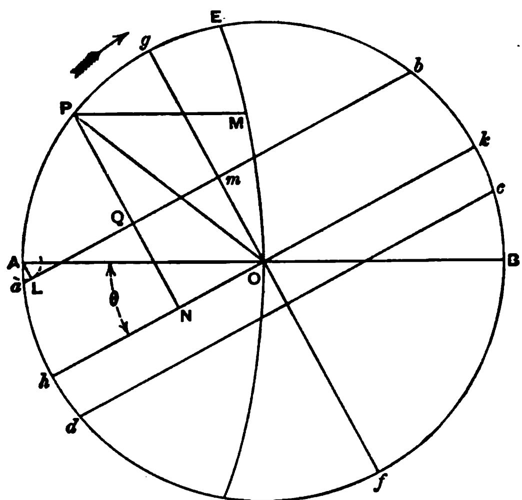


FIG. 84.

represent the eccentric's motion on one scale and the crank's on another, and let AB be the piston stroke. Draw hk as before so that the angle $AOh = \theta$, the angular advance. Taking a centre

on OA produced, draw the arc EOF through the centre O with radius equal to the length of the connecting-rod. Then when the crank has any position OP the displacement of the valve from its middle position is PN (drawn perpendicular to hk) and the displacement of the piston from mid-stroke is PM. Also, if ab and cd be drawn as before at distances from hk equal to the laps, the four events happen at a, b, c, and d, and PQ is the extent to which the valve is open when the crank is at P. Similarly AL is the extent to which the valve is open at the beginning of the stroke, that is the *lead*. The port has its maximum opening when the crank is at Og during admission and at Of during exhaust, unless its width is so small that it has become completely uncovered with a smaller displacement of the valve.

The diagram shown in fig. 84, which is a modified form of one due to Reuleaux, may readily be applied to determine the characteristics which a slide-valve must have to give a stated distribution of steam. Suppose for instance that the travel of the valve, the lead, and the position of cut-off are assigned. Having marked b, the position of the crank-pin at the given point of cut-off in relation to the stroke AB, draw a circle with centre A and radius AL equal to the lead. Then draw a line through b tangent to this circle. This will be the line ba of the diagram. Its inclination to BA determines the angular advance, and a perpendicular on it from O gives Om, which is the outside lap. The inside lap becomes determinate when either the point of release c or that of compression d is assigned, and it is found by drawing a line through c or d parallel to ab, and measuring the distance of this line from O.

159. Inequality of the distribution on the two sides of the piston. So far we have dealt only with the events corresponding to one end of the cylinder, namely (in the diagram) the back end. This has been done only to avoid complicating the diagram with too many lines. In fig. 85 the construction of fig. 84 is repeated with the outside-lap lines ab and $a'b'$ drawn for both ends, and also the inside-lap lines cd and $c'd'$, and the corresponding events are marked. The construction lines relating to the front end of the cylinder are distinguished by being dotted and their reference letters are accented. The laps have been taken equal for the two ends, and an obvious result is that the cut-off is considerably later at the back than at the front.

Compare bm with $b'm'$, these being the distances by which the piston has passed mid-stroke when the cut-off occurs at

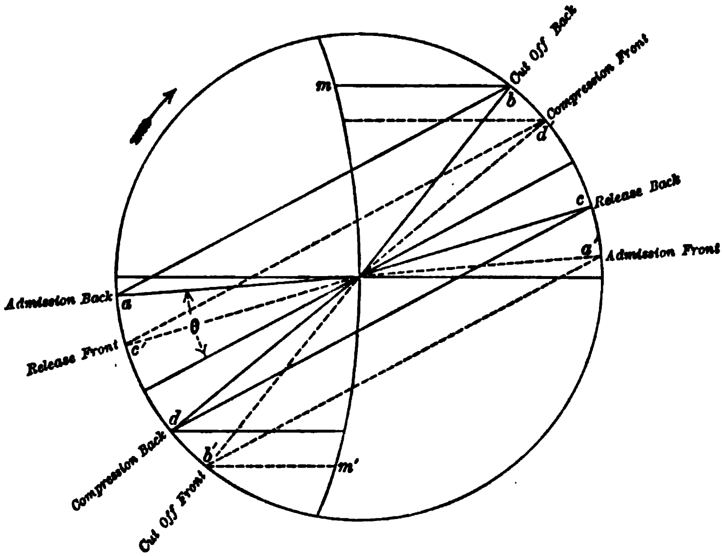


FIG. 85.

the back and front respectively. This want of symmetry, proceeding as it does from the obliquity of the connecting-rod, is slight when the rod is many times longer than the crank but becomes important when the rod is short. In the sketches the length of the rod is supposed to be three times that of the crank.

This inequality may be remedied by making the outside laps unequal, giving less lap to the front end of the valve. When this is done, however, the amounts of lead (which are equal with equal laps) become unequal. But in general it is better to sacrifice equality of lead and to secure at least approximate symmetry in the positions of the two points cut-off, when the admission of steam is controlled by a simple slide-valve. When a separate expansion valve is used (§ 168 below) the cut-off is determined by it, and not by the main slide-valve, and in that case the amounts of lead may properly be made equal. In certain cases a somewhat unequal distribution of steam is to be preferred, as in the ordinary vertical marine engine, where the work done by the steam against the front or bottom end of the piston is partly spent in raising the piston and rods and consequently should be greater than the

work done against the back or top end which is supplemented by the descent of these heavy weights. This difference may be allowed for by providing a later cut-off at the front than at the back, which is done by making the laps still more unequal than a symmetrical distribution would require.

In cases where the eccentric-rod is itself so short that its obliquity should be taken account of, this is readily done in Reuleaux's diagram (fig. 84 or 85) by using circular arcs in place of the straight lines *ab*, *hk*, *cd*, these arcs being described with a radius which represents the length of the eccentric-rod on the same scale as that on which the diameter AB represents the travel of the valve, from centres on Of produced beyond f. Except in rare cases it leads to no appreciable error to treat the eccentric-rod as infinitely long.

Fig. 86 illustrates how a symmetrical distribution is secured by reducing the outside lap at the front end. There *ab* is

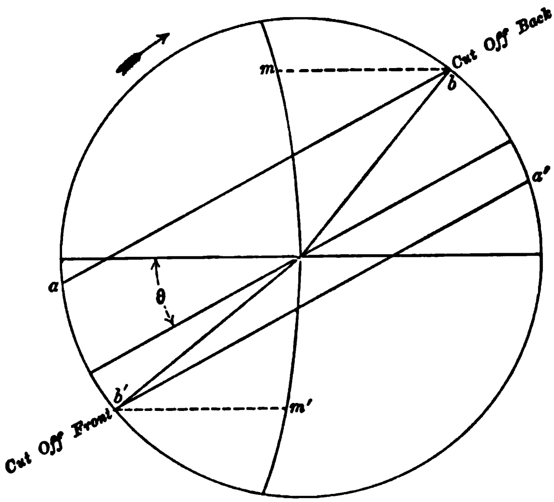


FIG. 86.

the outside-lap line for the back end and *a′b′* is the corresponding line for the front end. These lines are drawn so that the cut-off occurs at the same percentage of the stroke at both ends: *bm* and *b′m′* are equal. The inside laps may also be adjusted in the same way to give equal amounts of compression on both strokes (or, alternatively, to give symmetrical points of release). The amounts of lead, of course, are no longer equal: the lead at the

front end has been considerably increased by the reduction of the lap.

160. Zeuner's Valve Diagram. The graphic construction most usually employed in slide-valve investigations is the ingenious diagram published by Dr G. Zeuner in the *Civilingenieur* in 1856[1]. On the line AB (fig. 87), which represents the travel of the valve, let a pair of circles (called valve-circles) be drawn,

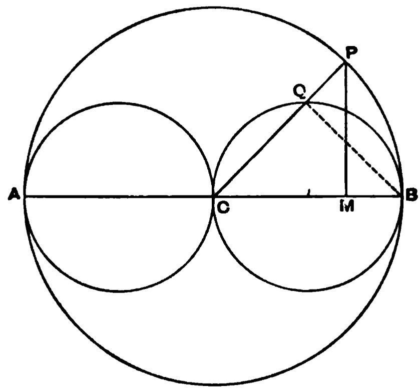


FIG. 87.

each with diameter equal to the half-travel. If a radius CP be drawn in the direction of the eccentric centre at any instant, it is cut by one of the circles at a point Q such that CQ represents the corresponding displacement of the valve from its middle position. That this is so will be seen by drawing PM and joining QB, when it is obvious that the triangles CPM and CBQ are equal in all respects and $CQ = CM$, which is the displacement of the valve. The line AB with the circles on it may now be turned back through an angle of $90° + \theta$ (θ being the angular advance), so that the valve-circles take the position shown to a larger scale in fig. 88. This makes the direction of CQP (the eccentric) coincide on the paper with the simultaneous direction of the crank, and hence to find the displacement of the valve at any position of the crank we have only to draw the line CQP in fig. 88 parallel to the direction which the crank has at the instant under consideration, when CQ represents the displacement of the valve to the scale on which the diameter of each valve-circle represents the

[1] Zeuner, *Treatise on Valve-Gears*, transl. by M. Müller, 1868.

half-travel of the valve. CL is the valve's displacement at the beginning of the stroke indicated by the arrow. Draw circular arcs

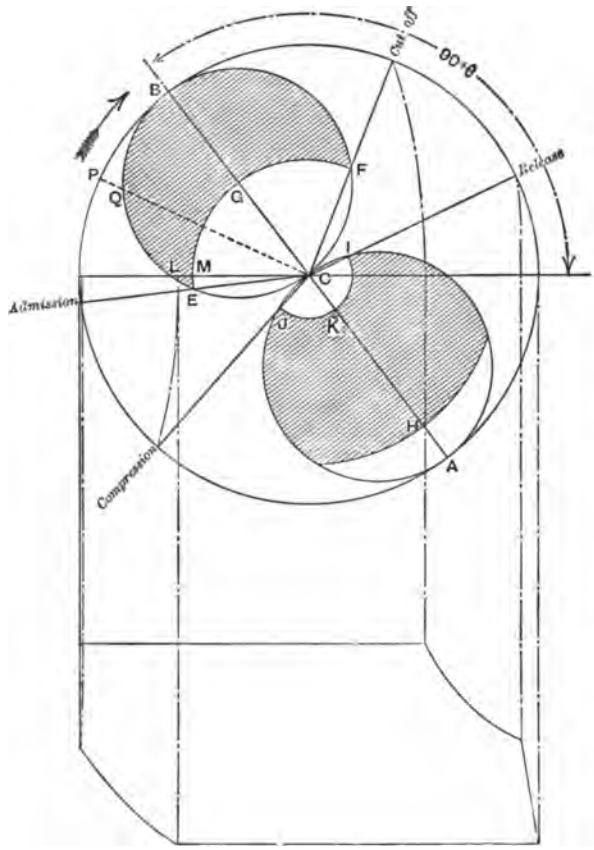


FIG. 88. Zeuner's Slide-Valve Diagram.

EF and IJ with C as centre and with radii equal to the outside lap and the inside lap respectively. CE is the position of the crank at which preadmission occurs. The lead is LM. The greatest steam opening during admission is GB, and the greatest opening to exhaust is the whole width of the port, namely KH. Intercepts on the radii within the shaded areas give the steam and exhaust openings for any angular positions of the crank. The cut-off occurs when the crank has the direction CF. CI is the position of the crank at release, and CJ marks the end of the exhaust, or the beginning of compression.

In the diagram given in fig. 88 radii drawn from C mark the angular positions of the crank, and their intercepts by the valve-

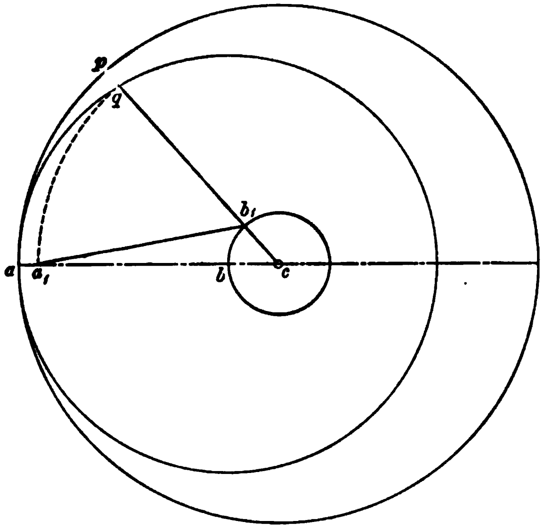


FIG. 89. Zeuner's construction to find the Displacement of the Piston.

circles determine the corresponding displacements of the valve. It remains to find the corresponding displacements of the piston. For this Zeuner employs a supplementary graphic construction, shown in fig. 89. Here ab or a_1b_1 represents the connecting rod, and bc or b_1c the crank. With centre c and radius ac a circle ap is drawn, and with centre b and radius ab another circle aq. Then for any position of the crank, as cb_1, the intercept pq between the circles is equal to aa_1, and is therefore the distance by which the piston has moved from the extreme position which it had at the beginning of the stroke. In practice this diagram is combined with that of fig. 88, by drawing both about the same centre and using different scales for valve and piston travel. A radial line drawn from the centre parallel to the crank in any position then shows the valve's displacement from its middle position by the intercept CQ of fig. 88, and the simultaneous displacement of the piston from the beginning of its motion by the intercept pq of fig. 89. As an alternative to this the piston's displacement may be found in Zeuner's diagram by the construction used in Reuleaux's, which was described in connexion with figs. 83 and 84.

As an example of the application of Zeuner's diagram we may take the same problem as before, namely, to find the outside lap and angular advance when the point of cut-off and the lead for the corresponding side of the piston are assigned, as well as the travel of the valve. The solution is shown in fig. 90. On the base-line

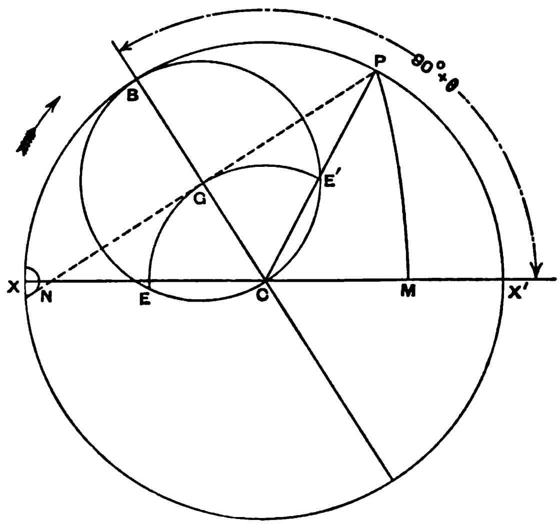


FIG. 90.

XX' mark the point M to represent the required point of cut-off and project this on the circle XPX' to find CP, which is the angular position of the crank at cut-off. With X as centre draw a circle with radius XN to represent the given lead. From P draw PN tangent to this circle. Then CGB drawn perpendicular to PN defines the diameter of the valve-circle. The angle $X'CB$ is the angular advance, plus 90°, and CG is the required outside lap.

So far as the simpler problems of the slide-valve are concerned Zeuner's diagram has no marked advantage over Reuleaux's. It is however more readily applicable to cases where the events of the stroke depend on the movements of more than one eccentric, as, for instance, in the Meyer expansion gear to be presently described.

161. Bilgram's Valve Diagram. This is an alternative construction shown in fig. 92 and arrived at as follows. In fig. 91 let AED be a circle representing the path of the eccentric-centre E, and let C be the angular position of the crank when

the eccentric is at E, so that COE is $90° + \theta$. Let OD be drawn making the angle AOD equal to the angular advance θ. Draw

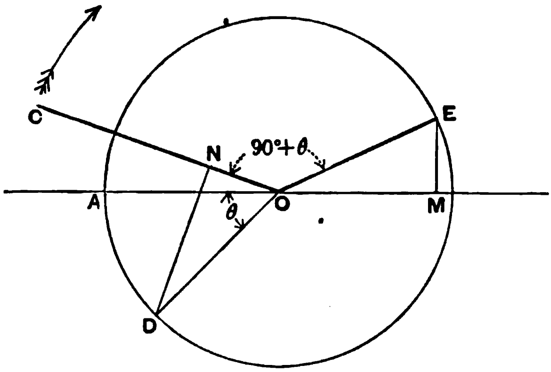


FIG. 91.

DN perpendicular to the direction of the crank OC. Then the triangles ODN and OEM are equal, and DN, being equal to OM, measures the displacement of the valve from its middle position when the crank is at C.

Hence the position of the crank when admission takes place is determined by the fact that the perpendicular from D on OC must then be equal to the outside lap: in other words, OC will then be tangent to a circle described about D as centre with the outside lap as radius. Further, when the crank travels round so that CO produced is tangent to the same circle on the other side it has reached the position of cut-off. Similarly, if a second circle be drawn about D with the inside lap as radius, the positions in which OC is tangent to it on its two sides are those of compression and release.

The complete construction, for one end of the cylinder, is shown in fig. 92. There AOD is the angular advance, OD the half-travel of the valve, DP the outside lap, and DQ the inside lap. C_1, C_2, C_3, C_4 are the positions of the crank at admission, cut-off, release, and compression, respectively. OL, being the half travel minus the outside lap, is the greatest opening of the port to steam. DF, drawn perpendicular to AO, is the displacement of the valve when the piston is at the dead-point A, and FG, the intercept on this line between AO and a line GH drawn parallel to AO touching the outside-lap circle, is the lead.

In illustration of the use of this diagram we may take two problems. (1) Given the travel of the valve, the position of the crank

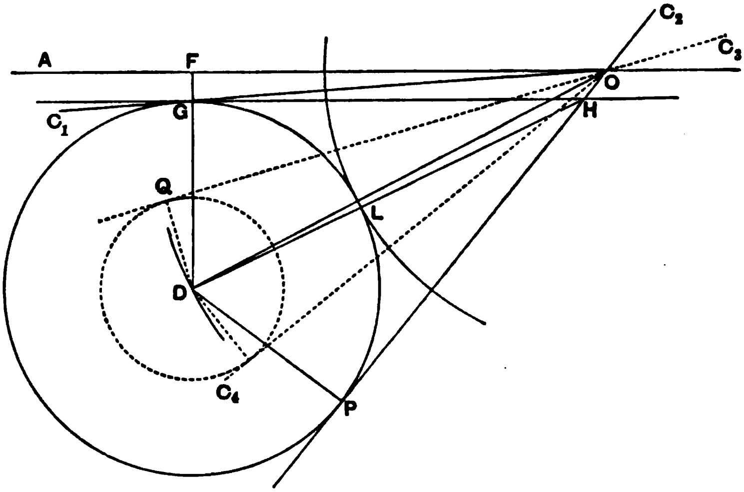


FIG. 92.

at cut-off, and the lead; to find the lap and the angular advance. Taking a base AO to represent the direction of motion of the piston, draw OC_2 parallel to the crank at cut-off, and produce it through O towards P. Draw a line GH parallel to AO and at a distance from it equal to the given lead. Bisect the angle GHP. The centre of the lap circle is on this bisecting line, and its distance from H is found by sweeping an arc with centre O and radius OD equal to the half travel. When D is found the circle is drawn touching GH and PH, and its radius gives the required lap. By joining D with O we have the angle AOD, which gives the angular advance.

(2) Given the position of the crank at cut-off, the lead, and the greatest opening of the port to steam; to find the lap and the angular advance, and the travel of the valve. Draw OC_2 as before to represent the position of the crank at cut-off and produce it through O. Draw GH parallel to AO at a distance below it equal to the lead: with centre O and radius OL equal to the given greatest opening of the port draw an arc L. Then find a centre D such that a circle may be drawn touching this arc and the lines GH and OP: this is most conveniently done by trial. The radius

of the circle gives the lap and AOD gives the angular advance as before.

162. Oval Diagram. A diagram is sometimes drawn which represents by a single curve the simultaneous displacements of the piston and the valve. When the position of the valve has been determined at various phases of the piston's stroke, whether by Reuleaux's or Zeuner's or any other method, a curve is drawn having for ordinates the displacement of the valve, on a base AB (fig. 93) which is the stroke of the piston, the scale of the ordinates being suitably exaggerated to prevent the curve from being inconveniently flat on account of the comparatively small amplitude of the valve's motion. This gives a species of oval figure resembling an ellipse, but somewhat distorted through the influence of the connecting-rod's obliquity. To find the events of the distribution, lines EE and II are drawn above and below the base at distances from it equal to the outside and inside laps respectively; their points of intersection with the curve at a, b, c and d mark the four events for the corresponding end of the cylinder. For the other end the outside-lap line $E'E'$ is to be drawn below the

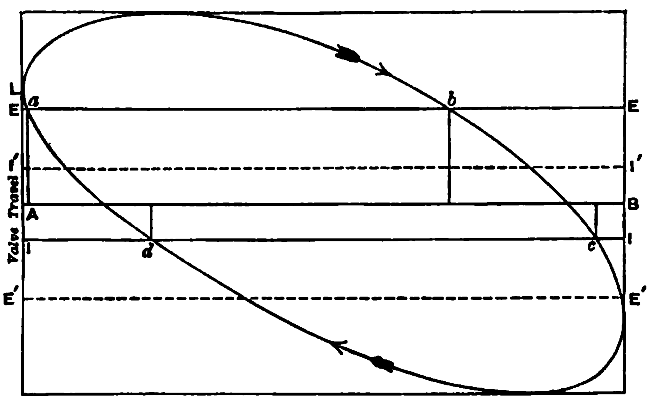


FIG. 93. Oval Diagram for the Slide-Valve.

base and the inside-lap line $I'I'$ above it. The distance of the curve beyond the outside-lap line shows at any stage in the stroke the extent to which the steam port is then open. The lead, which is EL, is not well defined in this form of graphic construction.

163. Harmonic Diagram. A much more useful diagram is obtained by drawing (preferably on section paper) separate

curves to represent the displacements of piston and valve respectively, each in relation to the angle turned through by the crank-shaft. Taking a base (fig. 94) the length of which repre-

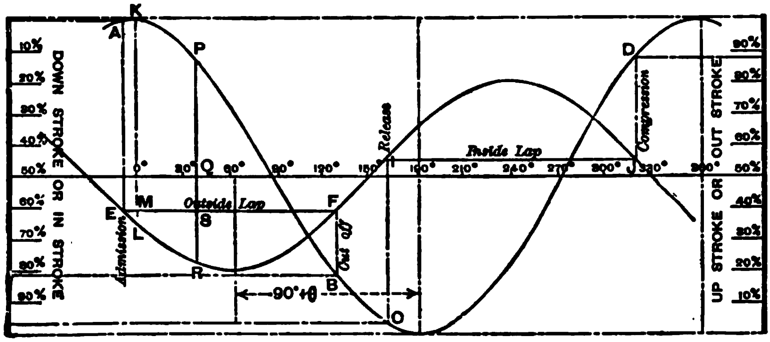


FIG. 94. Harmonic Diagram for the Slide-Valve.

sents the angle turned through in one revolution, let the curve *ABCD* be drawn to represent by its ordinates the displacement of the piston from mid-stroke, for all positions of the crank. Similarly let a curve *EFIJ* be drawn with ordinates which are (on any conveniently exaggerated scale) the displacements of the valve. Owing to the angle between the crank and the eccentric the phase of this curve is $90° + \theta$ in advance of the other: in other words, the valve attains its maximum displacement at a point on the base-line $90° + \theta$ earlier than (or to the left of) the point at which the piston attains its maximum displacement towards the same side. In drawing fig. 94 an angular advance of 30° has been assumed, which makes the total displacement of the valve curve to the left correspond to 120°. When questions have to be considered regarding the effect of varying the angular advance, one or other of the curves should be drawn on tracing paper in order that it may readily be slipped over the other into the position that will correspond to any desired angle.

Let any line *PQR* be drawn perpendicular to the base-line to intersect the piston curve in *P* and the valve curve in *R*. The displacement of the piston is then *PQ* and that of the valve is (on another scale) *QR*. The position of the piston in its stroke is found by projecting *P* upon the end line of the diagram (to the left) where a scale is marked to show percentages of the stroke. If *EF* be drawn parallel to the base and at a distance below it equal to the outside lap, *SR*, which is the excess of the valve's displacement

beyond the lap QS, gives the steam opening at the same phase of the stroke. Admission begins at E, and the corresponding position of the piston is found by projecting E upon the piston curve at A and then projecting A upon the scale at the side. The vertical distance from K to A shows the amount of preadmission. At K, the dead-point of the crank, the valve is open to the extent LM; in other words, LM is the lead. Cut-off occurs at F, and the corresponding position of the piston is found by projecting F upon the piston curve at B, and then projecting B upon the scale at the side. In the same way the positions of the piston at release and compression correspond to the points I and J on the valve curve when the line IJ is drawn at a distance above the base equal to the inside lap. All these events relate to one side of the piston; to obtain the events for the other side the outside-lap line has to be drawn above the base and the inside-lap line below it, and the points found on the piston curve are to be projected upon the scale which is set out on the right-hand side of the diagram in fig. 94. The inequality of lap and lead which is needed to give a symmetrical distribution, and other such problems of design, may be studied by help of this diagram with great ease and clearness[1]. Another example of its use will be given below in connexion with separate expansion valves (§ 169).

The ordinates of these curves may be found either by graphic construction or by calculation. As to the valve curve, the length of the eccentric rod is generally so great that its influence may be neglected, and in that case the formula

$$y' = r' \cos a'$$

may be used, r' being the eccentricity or the half-travel of the valve and a' being the angle through which the eccentric has turned from the position that corresponds to the maximum displacement of the valve.

In the piston curve the influence of the rod is usually considerable. Let r be the effective length of the crank AP (fig. 95)

[1] The writer is indebted to Professor O. Reynolds for drawing his attention to the advantages of the construction illustrated in fig. 94. As a means of solving slide-valve problems it is in several ways superior to the methods more generally used by draughtsmen. The labour of drawing the curves is considerable; but a set of curves drawn once for all, for various ratios of crank to connecting rod, will allow any particular case to be dealt with very readily. See a paper by Prof. W. E. Dalby, *Engineering*, April 7, 1893.

and l that of the connecting rod BP; when the crank has turned

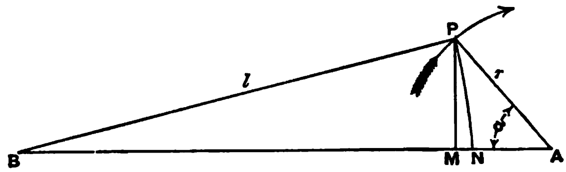


FIG. 95.

through any angle a from the dead-point the displacement of the piston from its middle position is

$$y = AN = AM + MB - l$$
$$= r\cos a + \sqrt{l^2 - r^2\sin^2 a} - l,$$

or, writing μ for the ratio of the length of the connecting rod to that of the crank,

$$y = r(\cos a + \sqrt{\mu^2 - \sin^2 a} - \mu).$$

This is always less than $r\cos a$, but approximates closely to that when μ is very great. An expression of the same form is of course applicable to the displacement of the valve and should be used when the eccentric rod is so short as to require its length to be taken into account. The angles a (for the crank) and a' (for the valve) are connected by the equation $a' = a + 90° + \theta$ where θ is the angular advance as before.

164. Reversing Gear. The Link-motion. In locomotives, marine engines, winding engines, traction engines and some other types it is necessary to make provision for reversing the direction in which the engine runs. A primitive way of doing this is to shift the eccentric of the slide-valve round upon the shaft until it takes relatively to the crank the angular position proper to the reversed motion. The eccentric must stand in advance of the crank by an angle equal to $90° + \theta$, and if its position be CE (fig. 96) while the crank is at CK the engine will run in the direction of the arrow A. To set the engine in gear to run in the opposite direction it is only necessary to shift the eccentric into the position CE', when it will still be in advance of the crank by the proper angle, the

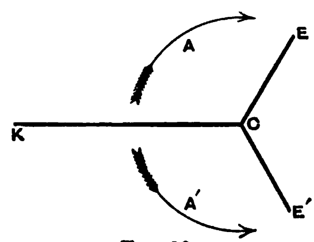


FIG. 96.

direction of motion now being that shown by the arrow A'. In some of the older engines this was substantially the actual method of reversal. The valve rod was temporarily disengaged from the eccentric and the valve was moved by hand in such a way as to make the engine begin to turn backwards. It was allowed to turn until the crank had moved back through an angle equal to ECE', the eccentric meanwhile remaining at rest, and the valve rod was then re-engaged. To allow the eccentric to remain at rest while the crank turned back through the required angle, the eccentric sheave instead of being keyed to the shaft fitted loosely on it and was driven by means of a spur fixed to the shaft which abutted on one or other of two stops or shoulders projecting from the sheave. Consequently when the engine shaft began to turn backwards the eccentric sheave did not at once follow it, until it had turned through an angle corresponding to the distance between the two stops. This device of the loose eccentric is not entirely obsolete[1], but nearly all modern engines which require reversing gear use either the *link-motion* or one of the forms of *radial gear* to be presently described.

In the link-motion there are two eccentrics keyed to the shaft in positions which correspond to CE and CE' in fig. 96, and the ends of their rods are connected to the ends of a link which gives its name to the contrivance. In Stephenson's link-motion—the earliest and still the most usual form—the link is a slotted bar or pair of bars forming a circular arc with radius equal or nearly equal to the length of the eccentric rods (fig. 97), and

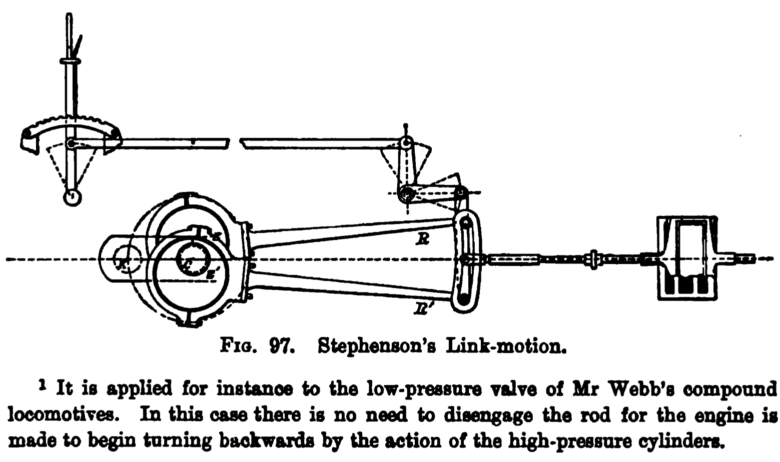


FIG. 97. Stephenson's Link-motion.

[1] It is applied for instance to the low-pressure valve of Mr Webb's compound locomotives. In this case there is no need to disengage the rod for the engine is made to begin turning backwards by the action of the high-pressure cylinders.

capable of being shifted up and down by means of a pendulum rod to which it is jointed either at one end or at the middle of the link. This suspension by a pendulum rod also allows the link to move sideways as the eccentrics revolve.

The valve-rod ends in a block which slides within the link, and when the link is placed so that this block is nearly in line with the forward eccentric rod (*R*, fig. 97) the valve moves in nearly the same way as if it were driven directly by a single eccentric. This is the position in "full forward gear." In "full backward gear," on the other hand, the link is pulled up until the block is nearly in line with the backward eccentric rod *R′*. The link-motion thus gives a ready means of reversing the engine,—but it does more than this. By setting the link in an intermediate position the valve receives a motion nearly the same as that which would be given by an eccentric of shorter throw and of greater angular advance, and the effect is to give a distribution of steam in which the cut-off is earlier than in full gear, and the expansion and compression are greater. Hence the mechanism also serves to adjust the amount of work done in the cylinder to the demand which may at any time be made upon the engine. In mid gear, which is the position sketched in the diagram, the steam distribution is such that scarcely any work is done in the cylinder. The movement of the link is effected by a hand lever, or by a screw, or (in large engines) by an auxiliary steam-engine. A usual arrangement of hand lever, sketched in fig. 97, has given rise to the phrase "notching up," to describe the setting of the link to give a greater degree of expansion, by bringing it nearer to mid gear. The eccentric rods are sometimes crossed instead of being "open" as shown in the sketch.

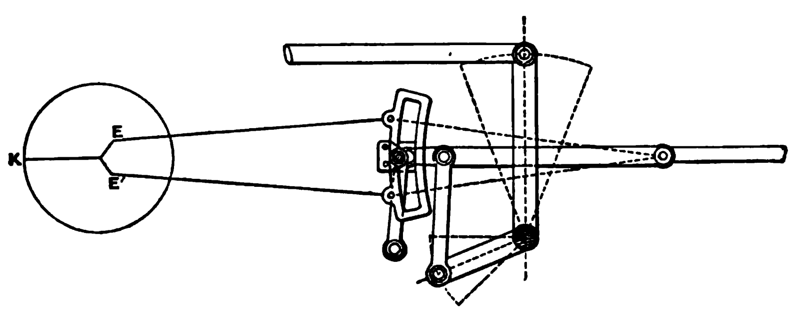


Fig. 98. Gooch's Link-motion.

In Gooch's link-motion (fig. 98) the link is not moved up in shifting from forward to backward gear, but a radius rod between the valve-rod and the link (which is curved to suit the length of this radius rod) is raised or lowered—a plan which has the advantage that the lead is the same in all gears. In Allan's motion (fig. 99) the change of gear is effected partly by shifting the link and partly by shifting a radius rod. This allows the link to be

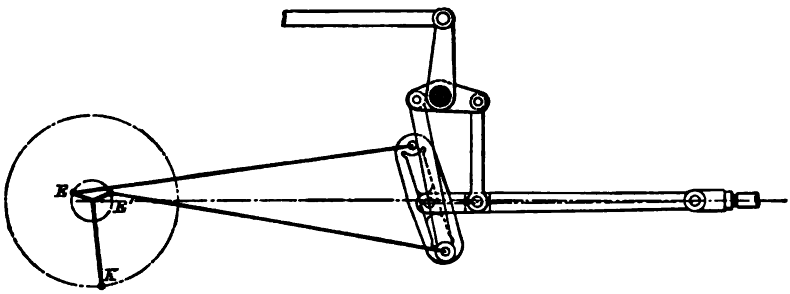


FIG. 99. Allan's Link-motion.

straight and has also the advantage that the weight of the link with its eccentric rod on the one hand, and the radius rod on the other, can be arranged to balance one another when a suitable proportion is given to the two arms of the short beam from which the pendulum rods hang.

165. Graphic Solution of the Link-motion. The movement of a valve driven by a link-motion may be fully and exactly investigated by drawing with the aid of a template the positions of the centre line of the link corresponding to a number of successive positions of the crank. Thus, in fig. 100, two circular arcs passing through e and e' are drawn with E and E' as centres and the eccentric rods as radii. These arcs are loci of two known points of the link, and a third locus is the circle a in which the point of suspension must lie. By placing on the paper a template of the link, with these three points marked on it, the position of the link is readily found, and by repeating the process for other positions of the eccentrics a diagram of positions (fig. 100) is drawn for the assigned state of the gear. A line AB drawn across this diagram in the path of the valve's travel determines the displacements of the valve, and enables the harmonic diagram to be drawn as in fig. 94, or alternatively the oval diagram as in fig. 93. The

example refers to Stephenson's link-motion in nearly full forward gear; with obvious modification the same method may be used

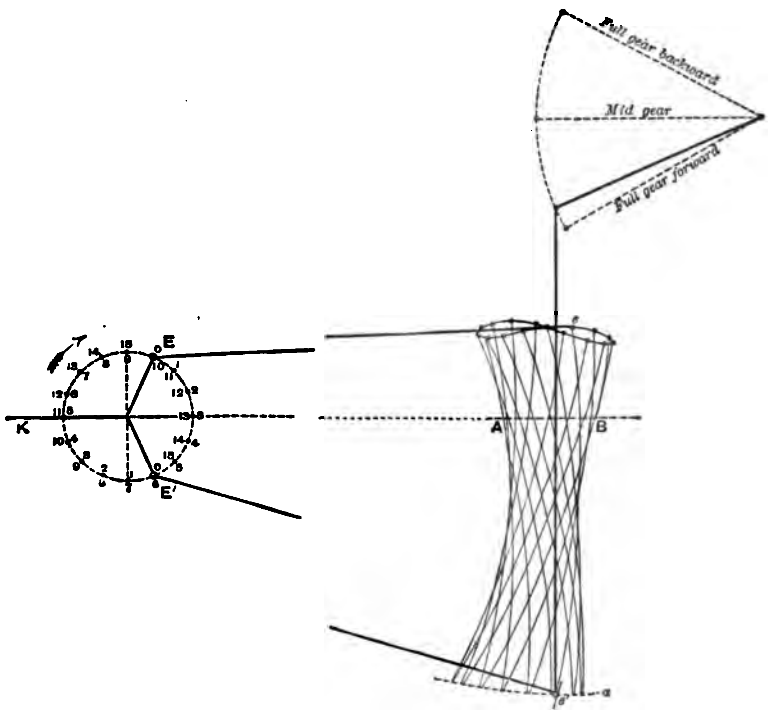


FIG. 100.

in the analysis of Gooch's or Allan's motion. The same diagram serves to determine the amount of slotting or sliding motion of the block in the link. In a well-designed gear this sliding is reduced to a minimum for that position of the gear in which the engine runs most usually. In marine engines the suspension-rod is generally connected to the link at that end of the link which is next the forward eccentric, in order to reduce this sliding as much as possible when the engine is running in its normal condition, namely, in forward gear.

166. Equivalent Eccentric. A less laborious, but less accurate, solution of link-motion problems is reached by the use of what is called the equivalent eccentric—an imaginary single eccentric, which would give the valve nearly the same motion as it gets from the link under the joint action of the two actual

eccentrics. The following rule for finding the equivalent eccentric, in any state of gear, is due to Mr MacFarlane Gray:—

Connect the eccentric centres E and E' (fig. 101) by a circular

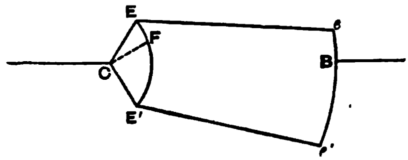


FIG. 101.

arc whose radius $=\dfrac{EE' \times \text{length of eccentric rod}}{2 \times ee'}$, ee' being the length of the link from rod to rod. Then, if the block is at any point B, take EF such that $EF : EE' :: eB : ee'$. CF then represents the equivalent eccentric both in radius and in angular position. If the rods of the link-motion are crossed instead of open,—an arrangement seldom used,—the arc EFE' is to be drawn convex towards C. Once the equivalent eccentric has been found the movement of the valve may of course be determined by Zeuner's or any of the other methods already described[1]. The method of the equivalent eccentric should not be taken as giving more than a first approximation to the actual motion; for anything like a complete study of a link-motion the graphic method of § 165 or the use of a model is to be preferred.

167. Radial Gears. Many forms of gear for reversing and for varying expansion have been devised with the object of escaping the use of two eccentrics, and of obtaining a more perfect distribution of steam than the link-motion can in general be made to give. Hackworth's Radial gear, the parent of several others, has a single eccentric E (fig. 102) opposite the crank, with an eccentric rod EQ, whose mean position is perpendicular to the travel of the valve. This rod ends in a block Q, which slides on a fixed inclined guide-bar or link, and the valve-rod receives its motion through a connecting rod from an intermediate point P of the eccentric rod, the locus of which is an ellipse. To reverse the gear the path in which Q moves is tilted over to the position shown by the dotted lines, and intermediate inclinations give various degrees of

[1] Examples of the application of Zeuner's diagram to the link-motion will be found in his book on Valve-Gears, cited above.

expansion without altering the lead. The steam distribution is excellent, and the cut-off is sharper than in the usual link-motion,

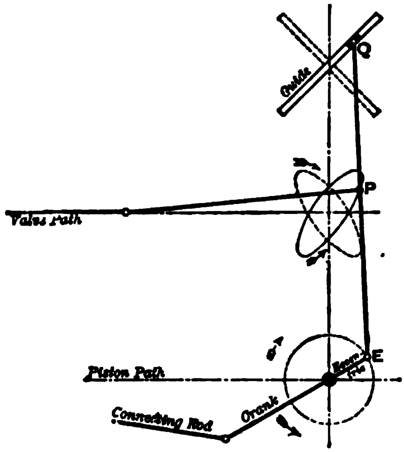


FIG. 102. Hackworth's Radial Valve-Gear.

but an objection to the gear is the wear of the sliding-block and guide. In a modified form of the Hackworth gear this objection is obviated with some loss of symmetry in the valve's motion by constraining the motion of the point Q, not by a sliding-guide as in fig. 102, but by a suspension-link, which makes the path of Q a circular arc instead of a straight line; to reverse the gear the centre of suspension R of this link is thrown over to the position R' (fig. 103). The same figure (103) shows another modification of what is substantially the same gear, namely, the placing of P beyond Q, with no angle between the crank and the eccentric; but P may be between Q and the crank (as in fig. 102), in which case the eccentric is set

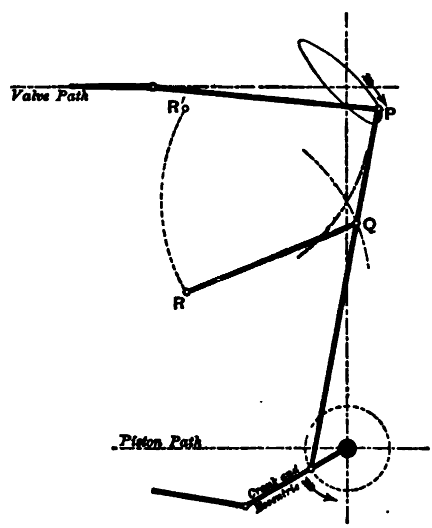


FIG. 103.

at 180° from the crank. This gear, as arranged by Bremme, Marshall and others, has been applied in a number of marine engines. Another type of radial gear is Joy's, which is extensively used in locomotives. In Joy's gear no eccentric is required; and the rod corresponding to the eccentric rod in Hackworth's gear receives its motion from a point in the connecting rod by the linkage shown in fig. 104, and is either suspended by a rod

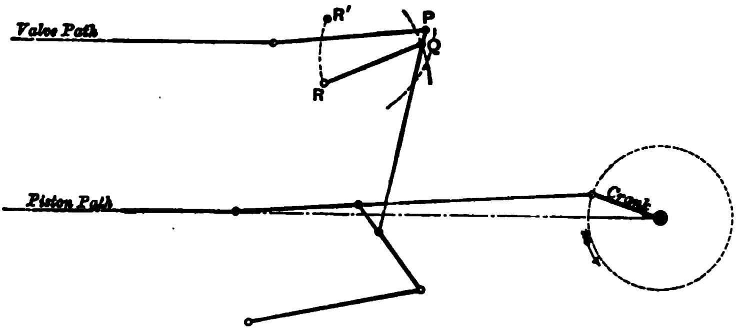


FIG. 104. Diagram of Joy's Valve-Gear.

whose suspension centre R is thrown over to reverse the motion, or constrained, as in the original form of the Hackworth gear, by a slot-guide whose inclination is reversed. Fig. 105 shows Joy's gear as applied to a locomotive. A slot-guide E is used, and it is

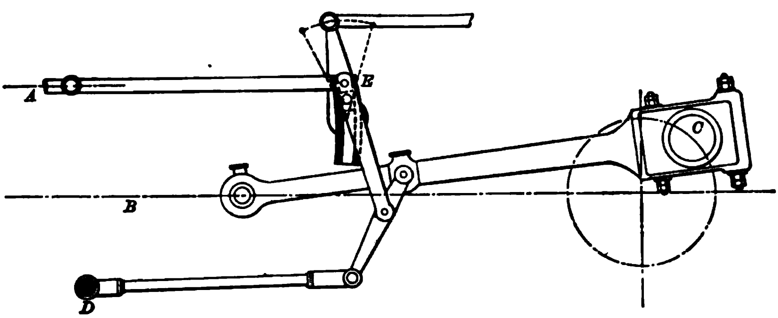


FIG. 105. Joy's Gear as applied to a Locomotive.

curved to allow for the obliquity of the valve connecting rod AE. C is the crank-pin, B the line of piston's motion, and D a fixed centre. Other forms of radial gear, dispensing with eccentrics and more or less closely related to the invention of Hackworth

have been designed by Walschaert, Morton, Brown, Bryce-Douglas, Kirk and others[1].

A method of reversing with a common slide-valve, which is used in steam steering engines[2], is to supply steam to what was (before reversal) the exhaust side of the valve and connect the exhaust to what was the steam side. This is done by means of a separate reversing valve through which the steam and exhaust pipes pass.

168. Separate Expansion-valves. When the distribution of steam is effected by the slide-valve alone the arc of the crank's motion during which compression occurs is equal to the arc during which expansion occurs, and for this reason the slide-valve would give an excessive amount of compression if it were made to cut off the supply of steam earlier than about half-stroke. Hence, when an early cut-off is wanted it is necessary either to use an entirely different means of regulating the distribution of steam, or to supplement the slide-valve by another valve,—called an expansion-valve, and usually driven by a separate eccentric,—whose function is to effect the cut-off, the other events being determined as usual by the slide-valve. Such expansion-valves belong generally to one or other of two types. In one, which is much the less common, the expansion-valve cuts off the supply of steam to the chest in which the main valve works. This may be done by means of a disk or double-beat valve (§ 174), or by means of a slide-valve working on a fixed seat (furnished with one or more ports) which forms the back or side of the main valve-chest. Valves of this last type are usually made in the "gridiron" or many-ported form to combine large steam-opening with small

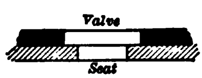


Fig. 106.

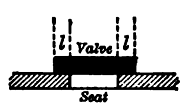


Fig. 107.

travel. Expansion-valves working on a fixed seat may be arranged so that the ports are either fully open (fig. 106) or closed (fig. 107)

[1] A discussion of Mr Joy's and other arrangements will be found in *Proc. Inst. Mech. Eng.* 1880. See also a paper by Mr J. Harrison on "Radial Valve-Gears," *Min. Proc. Inst. C. E.* Vol. cxiii., 1893.

[2] *Proc. Inst. Mech. Eng.* 1867.

when the valve is in its middle position. In the latter case the expansion-valve eccentric is set in line with or opposite to the crank, if the engine is to run in either direction with the same grade of expansion. Cut-off then occurs when the crank is at P (fig. 108), the expansion eccentric being at P', the shaft having

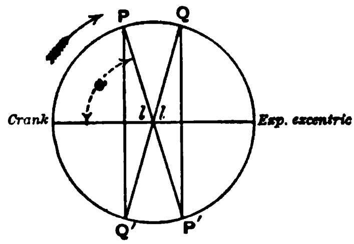


FIG. 108.

turned through an angle α from the beginning of the stroke. This is because the valve is then within a distance equal to l (fig. 107) of its middle position. The expansion-valve reopens when the crank is at Q, and the main slide-valve must therefore have enough lap to cut off earlier than $180°-\alpha$ from the beginning of the stroke, in order to prevent a second admission of steam to the cylinder. In the example shown in fig. 100 the expansion eccentric is set at right angles to the crank, which is a usual arrangement when the engine is provided with reversing gear, since it makes the cut-off happen at the same place in the stroke for both directions of running. If this condition need not be fulfilled, the expansion eccentric may have a somewhat different angular position, and in this way a more rapid travel at the instant of cut-off may be secured for one direction of running.

Since the separate expansion-valve of fig. 106 or 107 acts by cutting off the supply of steam from the steam-chest, but not directly from the cylinder, it does not prevent the steam which is stored in the chest from continuing to enter the cylinder until the main slide itself closes the admission port. When the cut-off by the expansion-valve is early and the steam-chest is capacious this affects the action materially.

169. **Meyer's Expansion-valve.** The other and much commoner type of expansion-valve is known as Meyer's. It consists of a pair of plates sliding on the back of the main slide-valve, which is provided with through ports which these

plates open and close. Fig. 109 shows one form of this type. Here it is the relative motion of the pair of plates forming the

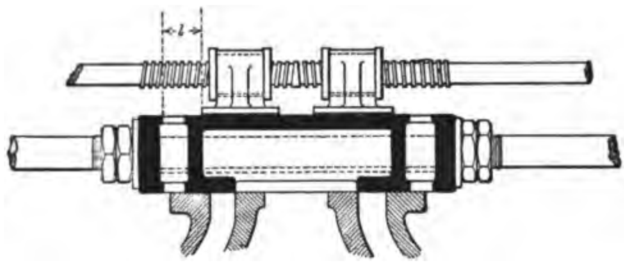


Fig. 109. Meyer Expansion-valve.

expansion-valve with respect to the main valve that has to be considered. If r_a and r_b (fig. 110) are the eccentrics working the main and expansion-valves respectively, then CR drawn equal and parallel to ME is the *resultant* eccentric which determines the motion of the expansion-valve relatively to the main valve. Cut-off occurs at Q, when the shaft has turned through an angle a, which brings the resultant eccentric into the direction CQ and makes the relative displacement of the two valves equal to the distance l. Another form of this valve (corresponding to the fixed-seat form shown in fig. 106) cuts off steam at the inside edges of the expansion-slides. With the form shown in fig. 109 the expansion eccentric will be set at 180° from the crank if the engine is to run in both directions with the same grade of expansion; otherwise a somewhat different angle may often be chosen with advantage, as giving a sharper cut-off.

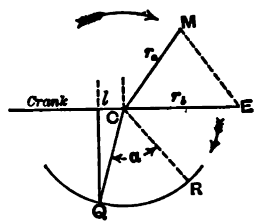


Fig. 110.

The action of Meyer's valve may be conveniently examined by the help either of Zeuner's diagram or of the harmonic diagram of § 163. Taking Zeuner's diagram first and assuming, for greater generality, that the expansion eccentric is not set just opposite the crank, let the circles I. and II. (fig. 111) be drawn to show, as in fig. 88, by the lengths of their chords through C the amount of absolute displacement of the expansion slide and main slide respectively each from its middle position, when the crank is in the angular position corresponding to the direction of the chord. In drawing these circles the angle XCM is set out to represent

the whole angle by which the main valve eccentric is ahead of the crank, as in fig. 88, and the angle XCE (also measured against the direction of the arrow) represents the angle by which the expansion eccentric is ahead of the crank, CM and CE being diameters of the circles II. and I. respectively. Then if any chord CQP be drawn from C to meet both circles the distance PQ, which is the difference between CP the absolute displacement of the main valve and CQ the absolute displacement of the expansion-valve, measures the relative displacement of one with respect to the other. This distance PQ is equal to the chord CR of a third circle (III.) drawn with CF equal and parallel to EM as its diameter. To prove this make the supplementary construction shown by the dotted lines. Then since the angles at P, Q, R and G are right angles, being angles in semicircles, PQ equals MG in

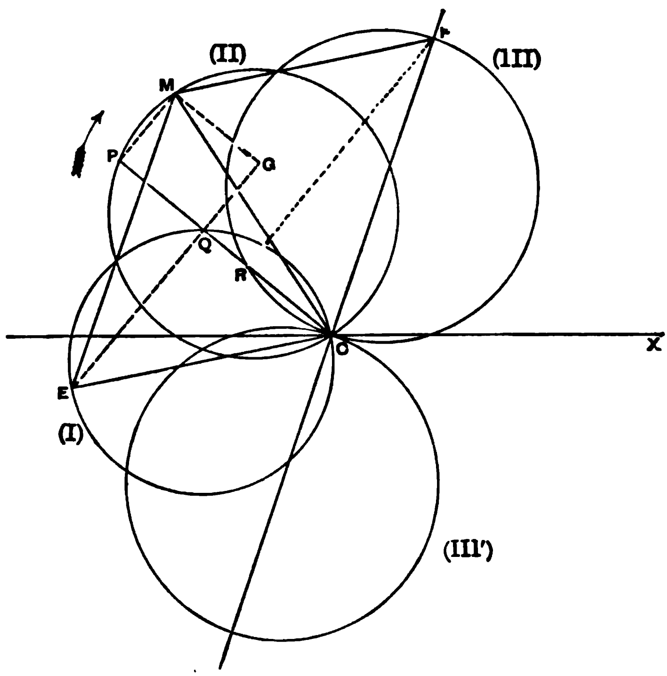


FIG. 111. Application of Zeuner's Diagram to the Meyer Expansion-valve.

the parallelogram PG, and RC equals MG in the equal triangles EMG and FCR. Hence PQ equals RC. A similar construction

of course applies to the return stroke, for which the circle showing the resultant motion is III′.

Thus by drawing the circles III. and III′. we at once determine, by the length of their chords through C, the relative displacement of the two valves for all positions of the crank. Cut-off, on the part of the expansion-valve, occurs when the crank is in such a position that the chord CR is equal to l (fig. 109), which is the amount of relative displacement that suffices to close the steam passage through the main slide. The expansion-valve reopens when the chord is again diminished to this value, towards the end of the stroke, and care must be taken that the main slide has enough outside lap to close the steam port leading into the cylinder before this stage in the revolution has been reached.

When the expansion-valve is furnished with a means of varying l, as in fig. 109, the point of cut-off may be made to take place early or late, the limit of earliness being imposed by the condition that l must not be reduced below the amount which will give a fair steam opening, and the limit of lateness being imposed by the consideration that the main slide itself becomes closed at a position determined by its own outside lap. The events of release, compression and admission, depending as they do on the main slide-valve alone are found by drawing lap arcs on the main-valve circle I. in the same manner as in earlier examples of Zeuner's diagram.

The harmonic diagram of § 163 gives an excellent means of studying the action of Meyer's valve. Three distinct curves having been drawn for the piston, main-valve and expansion-valve respectively, showing the displacement of each in relation to the angle turned through by the crank-shaft, they are to be superposed as in fig. 112 (using tracing paper as before) with the proper differences of angular position set out by distances measured along the base-line between the points at which the maximum displacement towards one side occurs in each. Both valve curves must have the same scale. Then the relative displacement of the valves is everywhere shown by the vertical distance between the main-valve curve and the expansion-valve curve. Cut-off is made to occur at any desired place in the motion by making the quantity l of fig. 109 equal to the distance found by measurement between the two valve curves at the corresponding point of the base-line. Thus in fig. 112, if it is

wished to cut off steam when the piston has travelled 25°/。 of its stroke, the corresponding point *P* is found by projection from the

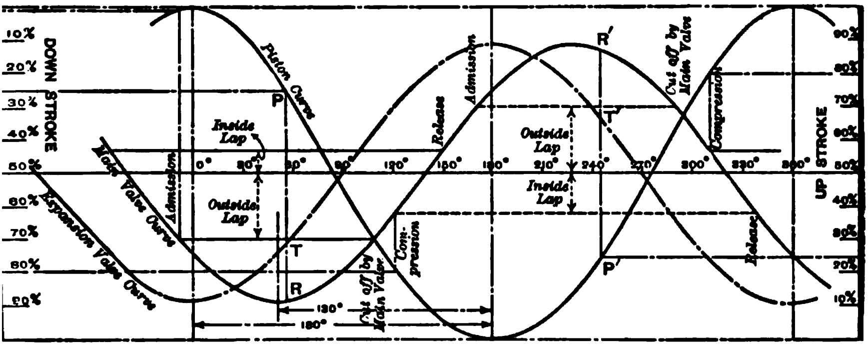


FIG. 112.

scale at the side, *PR* is drawn and the intercept *TR* is measured: this determines the proper length of the 'lap' *l*. With a smaller 'lap' the cut-off comes earlier, and in the particular example shown in the figure the admission may be reduced to 10°/。 of the stroke or even less by reducing *l*. The diagram[1] relates to a case in which the expansion eccentric was set at 180° in advance of the crank and the main-valve eccentric at 130° (making $\theta = 40°$). Both eccentrics had the same throw, giving a travel of 1·55 inches to each valve. The main-valve had an outside lap of 0·4 inches on both sides: this gave equal amounts of lead, namely 0·1 inches, but would have made the cut-off unequal on the two sides, namely at 70°/。 of the in-stroke or down-stroke and at 62°/。 of the out-stroke or up-stroke, if the cut-off had depended on the main-valve. Since the cut-off is accomplished earlier, by means of the expansion-valve, this inequality does not matter. The inside laps of the main-valve were made unequal so that they should give the same compression on both sides, namely by stopping the exhaust at 80°/。 in each back-stroke; their values, found by projection from points at 80°/。 on the stroke scale, were 0·24 inches on the front or bottom side and 0·14 inches on the back or top side.

By measuring distances such as *TR* between the two valve curves it will be seen that equal cut-off on the two sides can only be secured by having different values of *l* at the two ends of the

[1] Drawn by Prof. Dalby for a small vertical experimental engine in the Engineering Laboratory at Cambridge.

valve. Thus TR is 0·33 inches and $T'R'$, which also corresponds to a 25 °/₀ cut-off, is 0·42 inches, or nearly one-tenth of an inch more. A constant difference between the values of l at the two ends is in fact preserved in Meyer's gear while the values of l are varied, and in this case the diagram shows that a constant difference of about one-tenth of an inch suffices to keep the points of cut-off practically symmetrical from say $10°/_0$ to $35°/_0$ of the stroke. When the cut-off is to be later than $35°/_0$ equality can only be preserved by reducing slightly the difference between the values of l. The difference between them can be varied in practice by shifting the expansion-valve bodily towards or from its eccentric, provided the valve-spindle in the eccentric rod be furnished with a screw coupling or other device which permits its length to be altered.

The alteration of the expansion by varying the 'lap' l is accomplished in the ordinary form of Meyer's valve in a way which will be evident on reference to fig. 109. The valve rod has right and left-handed screws on it working in nuts which control the longitudinal positions on the rod of the two blocks that make up the expansion-valve. Hence by rotating the rod the blocks are made to approach or recede from each other, thus increasing or reducing the lap l at each end, but leaving any difference between the laps at the two ends unchanged. Matters are generally arranged so that this adjustment can be made while the engine is running, by means of a sleeve and hand-wheel which are usually fitted on a prolongation of the valve rod through the back end of the steam-chest. The cut-off may also be varied by altering the travel of the expansion-valve, instead of its lap. In some examples of the Meyer gear the expansion is varied automatically to suit the varying load upon the engine, the governor being connected to the expansion-valve in such a way that either the lap or more commonly the travel is varied in response to variation in the speed. When the travel is to be altered a link, oscillating about a fixed centre, is interposed between the valve rod and the eccentric rod, and by sliding the end of the eccentric rod up or down in the link, the link is made to act as a lever of variable length.

In a modified form of this valve, known as Rider's, the expansion-valve is a species of piston working in a cylindrical hole bored out of the main-valve. The steam passages terminate in a pair of oblique slots within this hole, and the front and back

edges of the piston-shaped expansion-valve are also cut obliquely, with the result that when the valve is turned about its axis its edges approach or recede from the oblique slots which form the steam ports. This turning can be effected by the governor.

170. Forms of Slide-valves. Double-ported valve. Trick valve. In designing a slide-valve the breadth of the steam ports in the direction of the valve's motion is determined with reference to the volume of the exhaust steam to be discharged in a given time, the area of the ports being generally such that the mean velocity of the steam during discharge is less than 100 feet per second. The travel is made great enough to keep the cylinder port fully open during the greater part of the exhaust; for this purpose it is $2\frac{1}{2}$ or 3 times the breadth of the steam port. To facilitate the exit of steam the inside lap is always small, and is often wanting or even *negative*, especially in engines which are designed to run at a high speed. During admission the steam port is rarely quite uncovered when the valve reaches the end of its travel, particularly if the outside lap is large and the travel moderate. Large travel has the advantage of giving freer ingress and egress of steam, with more sharply-defined cut-off, compression, and release, but this advantage is secured at the cost of more work spent in moving the valve and more wear of the faces[1]. To lessen the necessary travel without reducing the area of steam ports, double-ported valves are often used, and occasionally there are even three ports at each end. An example of a double-ported valve is shown in fig. 114. Fig. 113 shows the Trick valve, a device which accomplishes the same purpose by giving simultaneous admission in two ways; steam enters directly past the outer edge, as in an ordinary slide-valve and at the same instant an opening at the other end of the valve is uncovered by passing beyond the edge of the raised seat on which the valve works. This gives a supplementary admission, to the same cylinder port, through a passage cast in the back of the valve itself.

FIG. 113.
Trick Valve.

[1] For an experimental investigation of the friction of locomotive slide-valves see a paper by Mr J. A. F. Aspinall, *Min. Proc. Inst. C. E.* Vol. xcv., 1888.

Incidentally, fig. 114 illustrates an arrangement that is usual in all heavy slide-valves whose travel is vertical—the *balance-piston,* which is pressed up by steam on its lower side and so

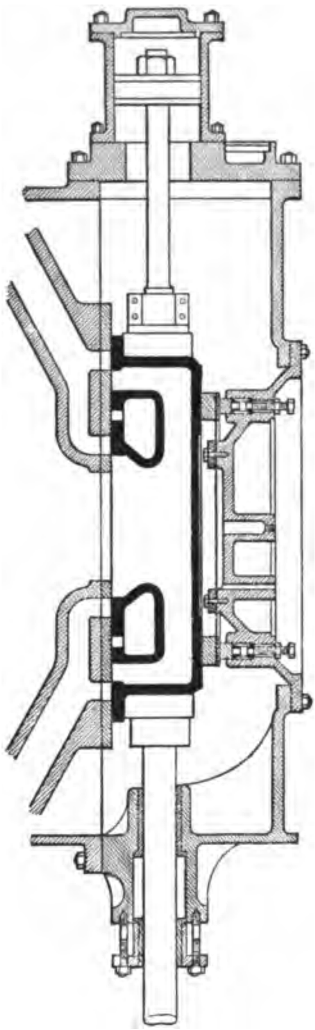


FIG. 114. Double-ported Valve with balance-piston and relief-frame.

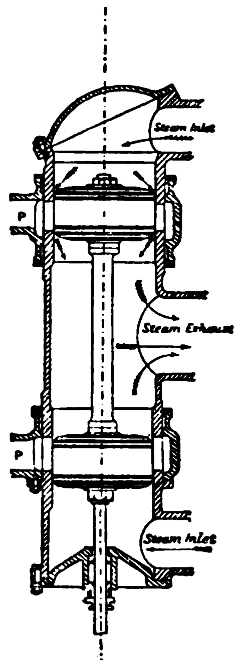


FIG. 115. Piston Slide-Valve.

equilibrates the weight of the valve, valve-rod, and connected parts of the mechanism.

171. Relief Frames. To relieve the pressure of the valve on the seat, large slide-valves are generally fitted with what is called a *relief-frame*, which excludes steam from the greater part of the back of the valve. In a common form of relief-frame a ring fits steam-tight into a recess in the cover of the steam-chest, and is pressed by springs against the back of the valve, which is planed smooth to slide under the ring. Another plan is to fit the ring into a recess on the back of the valve, and let it slide on the inside of the steam-chest cover. Steam is in either case excluded from the space within the ring, any steam that leaks in being allowed to escape to the condenser (or to the intermediate receiver when the arrangement is fitted to the high-pressure cylinder of a compound engine). A flexible diaphragm is sometimes used to make a steam-tight partition between the back of the relief-frame and the cover of the valve-chest, and in that case the frame may take the form of a rectangular casting with a planed face, which remains at rest while the valve, the back of which is also planed, slides beneath it. Fig. 114 gives an example of a relief-ring fitted on the back of a large double-ported slide-valve for a marine engine.

172. Piston Valves. The pressure of valves on cylinder faces is still more completely obviated by making the back of the valve similar to its face, and causing the back to slide in contact with the valve-chest cover, which has recesses corresponding to the cylinder ports and communicating with them. This arrangement is most perfectly carried out in the *piston slide-valves* now very largely used in the high-pressure cylinders of marine engines. The piston slide-valve may be described as a slide-valve in which the valve face is curved to form a complete cylinder, round whose whole circumference the ports extend. The pistons are packed like ordinary cylinder pistons by metallic rings, and the ports are crossed here and there by diagonal bars to keep the rings from springing out as the valve moves over them. Fig. 115 shows a form of piston valve for a marine engine. *PP* are the cylinder ports, and the supply of steam reaches the valve through two distinct inlets at the top and bottom. In another form of piston valve the rod connecting the two pistons is hollow and forms a communication between the steam chambers above and below the valve, thus making one steam inlet suffice.

An interesting variety of the piston valve occurs in the Willans 'central-valve' engine to be described in a later chapter. In this case the piston-rod of the engine is hollow and its interior forms the cylindrical chamber in which the valve slides, the events of the distribution being determined by the relative movement of the main piston-rod and the piston valve within it.

173. Rocking Slide-valve. The slide-valve sometimes takes the form of a disk revolving or oscillating on a fixed seat, and sometimes of a rocking cylinder (fig. 116). This last kind of sliding motion is very usual in stationary engines fitted with the Corliss gear, which will be more particularly described in the next chapter, in which case four distinct rocking slides are commonly employed to effect the steam distribution, one giving admission and one giving exhaust at each end of the cylinder (see fig. 138). A characteristic of the Corliss gear is that the steam valve after being held open during admission is disconnected from its eccentric by means of a "trip" device, which allows it to close suddenly under the action of a spring.

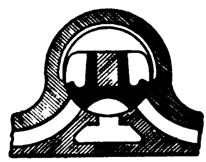


FIG. 116. Rocking Slide-Valve.

174. Double-beat valve. The Cornish cataract. In many stationary engines *lift* or *mushroom* valves are used, worked by tappets, cams, or eccentrics. Lift valves are generally of the

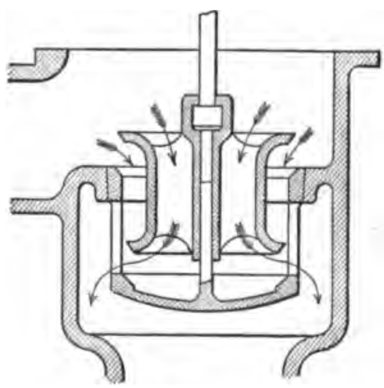


FIG. 117. Double-Beat Lift-Valve.

Cornish or double-beat type (fig. 117), in which equilibrium is secured or rather approximated to by the use of two conical

faces of nearly the same size which open or close together. In Cornish pumping engines, which retain the single action of Watt's early engine, three double-beat valves are used, as steam-valve, equilibrium-valve, and exhaust-valve respectively. These are closed by tappets on a rod moving with the beam, but are opened by means of a device called a cataract, which acts as follows. The cataract is a small pump with a weighted plunger, discharging fluid through a stop-cock which can be adjusted by hand when it is desired to alter the speed of the engine. The weighted plunger is raised by a rod which hangs from the beam, but is free in its descent, so that it comes down at a rate depending on the extent to which the stop-cock is opened. When it comes down a certain way it opens the steam and exhaust valves, by liberating catches which hold them closed; the "out-door" stroke then begins and admission continues until the steam-valve is closed; this is done directly by the motion of the beam, which also, at a later point in the stroke, closes the exhaust. Then the equilibrium-valve is opened, and the "in-door" stroke takes place, during which the plunger of the cataract is raised. When it is completed, the piston pauses until the cataract causes the steam-valve to open and the next "out-door" stroke then begins. By applying a cataract to the equilibrium-valve also, a pause is introduced at the end of the "out-door" stroke. Pauses have the advantage of giving the pump time to fill and of allowing the pump-valves to settle in their seats without shock.

The Sulzer engines, already referred to in chapter V. (§ 124), give one out of many examples that might be cited from Continental practice, in which the admission and exhaust are controlled by mushroom valves of the double-beat type. The exhaust-valves are placed below the (horizontal) cylinder and the admission valves are on the top. The latter are opened by eccentrics and are furnished with a trip gear which allows them to close suddenly, giving a sharp cut-off as in the Corliss engine[1].

[1] For other examples of mushroom valve-gears see Haeder's *Handbook of the Steam-Engine*, Tran. by H. H. P. Powles (1893).

CHAPTER IX.

GOVERNING.

175. Methods of regulating the work done in a Steam-engine. To make an engine run steadily an almost continuous process of adjustment must go on, by which the amount of work done by the steam in the cylinder is adapted to the amount of external work demanded of the engine. Even in cases where the demand for work is sensibly uniform, fluctuations in boiler-pressure still make regulation necessary. Generally the process of government aims at regularity of speed; occasionally, however, it is some other condition of running that is to be maintained constant, as when an engine driving a dynamo-electric machine is governed by an electric regulator to give a constant difference of potential between the brushes—a condition which often requires the engine to run rather faster when it is giving a greater output.

The ordinary methods of regulating are either (*a*) to alter the pressure at which steam is admitted by opening or closing more or less a throttle-valve between the boiler and the engine, or (*b*) to alter the volume of steam admitted to the cylinder by varying the point of cut-off. The former plan was introduced by Watt, and is still common, especially in small engines. From the point of view of heat economy it is somewhat wasteful, since the process of throttling is irreversible in the thermodynamic sense, but this objection is lessened by the fact that the wire-drawing of steam dries or superheats it, and consequently reduces the condensation which it suffers on coming into contact with the chilled cylinder walls. On the other hand, to hasten the cut-off does not involve a reduction of efficiency unless the ratio of expansion is already very great. The second plan of regulating is in general

to be preferred, especially when the engine is subject to large variations of load, and it is usually followed in stationary engines of the larger types.

176. Automatic regulation by Centrifugal speed Governors. Watt's Conical Pendulum Governor. Within certain limits regulation by either plan can be effected by hand, but for the finer adjustment of speed some form of automatic governor is necessary. Speed governors are commonly of the *centrifugal* type: a pair of masses revolving about a spindle which is driven by the engine are kept from flying out by a certain controlling force. When an increase of speed occurs this controlling force is no longer able to keep the masses revolving in their former path; they move out until the controlling force is sufficiently increased, and in moving out they act on the regulator of the engine, which may be a throttle-valve or some form of automatic gear by which the cut-off is varied. In the conical pendulum governor of Watt (fig. 118) the revolving masses are balls attached to a vertical spindle by rods, and the controlling force is furnished by the weight of the balls, which, in receding from the spindle, are obliged to rise. When the speed exceeds or falls short of its normal value they move out or in, and so raise or lower a collar C which is in connexion with the throttle-valve through a lever. The suspension-rods may be hung from the ends of a T-piece attached to the revolving spindle instead of being pivoted in the axis as in fig. 118, and in some cases they cross each other and the spindle itself as in fig. 124.

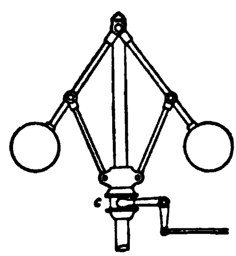
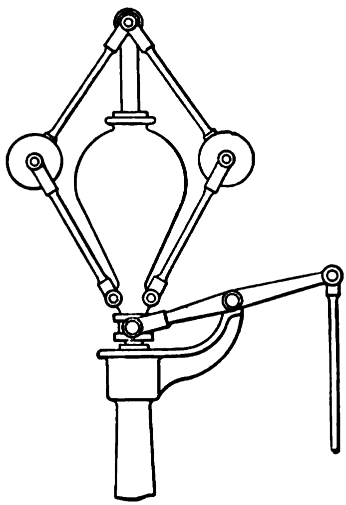


FIG. 118. Watt's Governor.

177. Loaded Governors. In a modified form of Watt's governor, known as Porter's, or the *loaded* governor, the tendency which the balls have to fly out is resisted not only by their own weight but also by a supplementary controlling force which is furnished by a weight resting on the sliding collar (fig. 119). This device is equivalent to increasing the *weight* of the balls without altering their *mass*. In other governors the controlling force instead of being due to gravity only is wholly or partly produced

by springs. Fig. 120 shows a governor by Messrs Tangye in



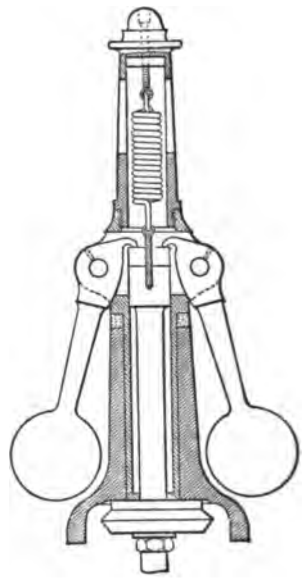


FIG. 119. Loaded Governor. FIG. 120. Spring Governor (Tangye).

which the balls are controlled partly by their own weight and

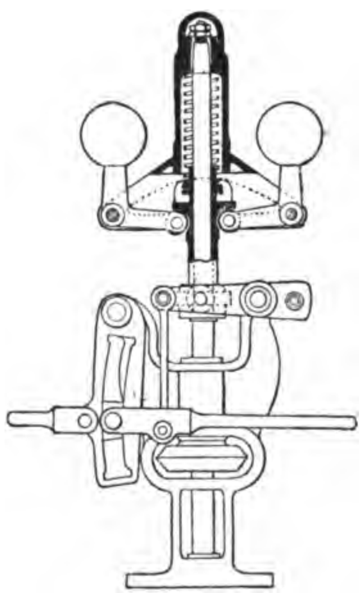


FIG. 121. Spring Governor (W. Hartnell).

partly by a spring, the tension of which is regulated by turning the cap at the top. Another example of a governor with spring control is shown in fig. 121. There the balls move in a sensibly horizontal line and consequently their weight contributes nothing towards resisting the tendency to fly out. A certain amount of additional control, however, is supplied by the weight of the sliding collar and the parts which rise with it, unless these are counterbalanced.

178. Controlling Force. In whatever way the tendency of the balls to fly out be resisted, whether through their own weight or through a supplementary load or through springs, it is convenient to treat the control as equivalent to a certain force F acting on each ball in the direction of the radius towards the axis of revolution. We shall call this the *controlling force*. The value of F varies, in a given governor, when the position of the balls changes. If it were not for friction, the controlling force for any position of the balls could be found experimentally by applying a spring-balance to each ball, with the governor at rest, and noting the force required to hold the ball in the assigned position when this force was applied directly away from the axis of the governor. Owing to friction such an experiment would give two extreme values of the force, for if the pull on the spring-balance were increased the ball would not move further out until the pull became equal to $F+f$, where f denotes the influence of friction. And if the effect of friction in resisting the return of the ball were the same as its effect in resisting the displacement outwards, the pull of the spring-balance might be reduced to $F-f$ before the ball would begin to move in. A mean of these extremes would give the true controlling force in cases where the influence of friction remained unchanged.

When the governor is running the influence of friction is in general less than when it is at rest; but the effect still is to make the actual force which the ball experiences, pulling it towards the spindle, greater or less than F according as the ball is on the point of moving out or moving in.

179. Condition of Equilibrium. Once the controlling force F is known the speed at which the governor must revolve in order to make the balls take up any assigned position is readily calculated. If M be the mass of the ball (in lbs.), n the number of

revolutions per second and r the radius (in feet) of the path in which the balls revolve, equilibrium will be maintained when

$$F = 4\pi^2 n^2 r M,$$

F, the controlling force, being expressed in poundals. Hence the speed corresponding to the assigned configuration of the governor is defined by the equation

$$n = \frac{1}{2\pi}\sqrt{\frac{F}{Mr}}.$$

For the present, friction is left out of account; its influence on the speed will be considered immediately.

180. Condition of Stability. A governor is said to be stable when (apart from the influence of friction) any small finite increase or decrease of speed above or below the speed proper to any given configuration makes the balls go out or come in by a finite amount, so that they reach a new position of equilibrium corresponding to the new speed.

In order that a governor should be stable F must increase more rapidly than r when the balls move outwards. That is to say, $\frac{\delta F}{F}$ must exceed $\frac{\delta r}{r}$. This follows, by the above equation, from the fact that the new position of the balls is to correspond to a greater value of the speed n.

If F varied just proportionally to r, n would be constant for all values of r. This state of things would correspond to neutral equilibrium on the part of the governor: its consequences are considered more particularly in §183 below.

181. Equilibrium of the Conical Pendulum Governor. Height of the Governor. When the governor is a simple

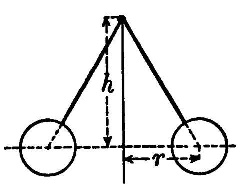


FIG. 122.

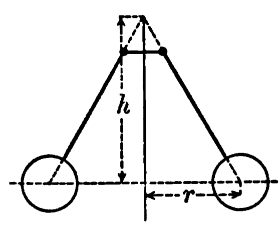


FIG. 123.

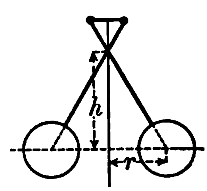


FIG. 124.

conical pendulum controlled by gravity only and without load other than the weight of the balls themselves—a condition never

quite realised in practice, since the weight of the sliding collar and its attached parts always applies some extra load which adds to the controlling force—F is the resultant of F_2 the tension in the suspending rod and F_1 or Mg the weight of the ball (figs. 122—124). The triangle of forces is sketched in fig. 125. This applies whichever of the three forms shown in figs. 122, 123, and 124 is given to the governor. Let the *height* of the pendulum governor, that is the vertical distance from the plane of rotation of the balls to the point where the axis of the suspending rod (produced if necessary) cuts the axis of the spindle, be called h. Then, using absolute units for the forces,

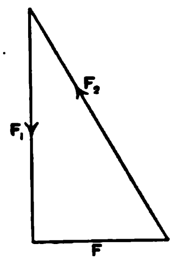


FIG. 125.

$$F : Mg :: r : h,$$

from which

$$F = \frac{Mgr}{h}, \text{ and } n = \frac{1}{2\pi}\sqrt{\frac{g}{h}}.$$

Hence in a governor of this type, the condition of stability may be expressed by saying that h must diminish when the speed increases. This will obviously happen if the form be that of fig. 122 or fig. 123. With the crossed-rod form of fig. 124 the height h will increase when the balls rise only if the points of suspension are not far from the axis. By placing them at a particular distance from the axis, h may be kept very nearly constant: in other words, this governor may be arranged to have nearly neutral equilibrium, so that a very small change in the speed n may be associated with a large change in the position of the balls. When the centres of suspension are put further from the axis the mechanism sketched in fig. 124 becomes unstable, and is then unfit to serve as a governor.

182. Equilibrium of Loaded Governor. The results obtained in the last paragraph are readily adapted to the case of a governor of the Porter type (fig. 119). Let M' be the amount of the extra load, per ball (in general M' is one-half the total extra load), and let q be the velocity ratio of the vertical movement of the load to the vertical movement of the ball—a quantity which is easily found by calculation or graphically when the form of the governor is given. Then each ball, in being displaced

outwards, has not merely to raise its own weight but has to raise what is equivalent to an additional weight equal to q times the weight of M'. The effect of the load is therefore to increase the controlling force F from $\frac{Mgr}{h}$, in poundals, to $\frac{(M+qM')gr}{h}$. But the condition of equilibrium still is that F should be equal to $4\pi^2n^2rM$. Hence the speed n at which the governor must now turn to maintain any assigned height h is

$$n=\frac{1}{2\pi}\sqrt{\frac{(M+qM')g}{Mh}}.$$

Compared with the simple or unloaded form, this governor requires a higher speed in the proportion of $\sqrt{M+qM'}$ to $\sqrt{M}$.

In the ordinary construction of the Porter governor the four links form a parallelogram and consequently the vertical movement of the load borne by the sliding collar is twice that of the balls, or $q=2$. And as the whole load is divided between two balls, each ball virtually has its weight, but not its mass, increased by an amount equal to the whole weight of the central load.

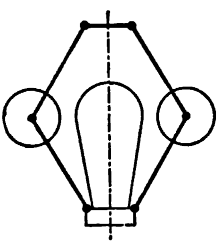


FIG. 126.

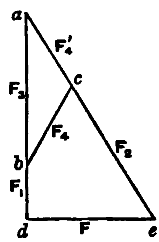


FIG. 127.

Another way of considering the equilibrium of the Porter governor may be mentioned. Let the mass of each ball be M and let that of the load be $2M'$ as before. The load, the weight of which is $2M'g$ in poundals, is borne by the tensions in the two lower rods (fig. 126). By drawing the triangle abc which is the diagram of forces for the load, and in which ab or F_3 is the weight of the load, we find the value of F_4 and F'_4 which are the tensions in the lower rods. Then draw bd or F_1 to represent the weight of one ball (namely Mg), and draw the horizontal line de to meet a line ce drawn from c parallel to the direction of

the upper rod. The figure *ecbd* is the polygon of forces acting on the ball, *ed* being the resultant controlling force F. The speed at which the governor will run is determined by the condition that $4\pi^2n^2rM$ is to be equal to this force *ed*. In the usual case of parallel rods, *ace* is one straight line and then

$$ed = ad \tan a, \text{ or } F = (M + 2M')g \tan a,$$

where a is the inclination of the rods to the vertical. Since $\tan a = \frac{r}{h}$ this expression agrees with the one given above.

183. Sensibility in a Governor. Isochronism. Any change of speed in a governor tends to produce a change in the position of the balls, and if the governor itself and the regulating mechanism connected with it were free from friction only one position of the governor would be possible for any one speed, provided the condition of stability were complied with. If therefore the supply of steam depends on the position taken up by the governor balls a stable governor does not maintain a strictly constant speed in the engine it controls. Whenever the boiler pressure or the demand for work changes a certain amount of displacement of the balls is necessary to increase or reduce the steam supply, and the balls can retain their new position only by virtue of continuing to turn slower or faster than before. The maximum change of speed which can occur under the control of the governor is that which will make the balls move from one to the other extremity of their range—namely, from the position which allows the full supply of steam to the position which completely checks the supply. Of course if the engine is overloaded by giving it too much external resistance to overcome, the speed may be further reduced after the governor has done all that it can do to let steam in freely, but the variation of speed for which the governor is responsible is only that which makes the change from no steam to full steam. When a small variation of speed suffices to do this the governor is said to be sensitive, its sensibility being measured by the reciprocal of the ratio which this variation of speed bears to the mean speed.

The more stable a governor is the less sensitive is it; on the other hand when the equilibrium is neutral the sensibility is indefinitely great. The controlling force F then varies as r,

and hence n is constant (§ 179) at whatever distance from the axis the balls revolve. In other words, the balls are in equilibrium at one speed and only at one (except for friction), and the least variation from this speed suffices to send them to one extremity or the other of their range. A governor having this quality is said to be isochronous. Friction makes the condition of strict isochronism impossible, but many governors are made nearly isochronous by arranging them so that, as the balls are displaced, the controlling force increases only a little more rapidly than r.

184. Isochronism in the Gravity Governor. Parabolic Governor. An ideal frictionless governor, in which the controlling force is furnished by gravity, can be made isochronous if the balls instead of being hung by rods from fixed points are constrained to move in a parabolic path, as in fig. 128, where the cup or channel which holds the ball is so shaped that the locus of

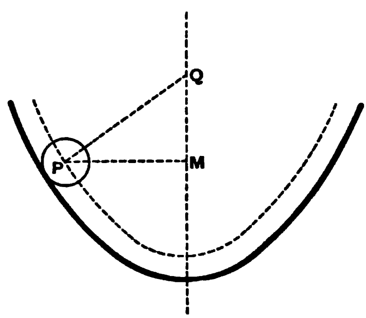


FIG. 128.

the centre of the ball, shown by a dotted curve, is a parabola. The pressure of the ball against the cup is equivalent to the tension of an imaginary suspension rod PQ; and it is a property of the parabola that the sub-normal QM, which represents h, is constant wherever P be taken along the curve. Hence a ball supported in this way would remain in equilibrium at one particular speed of rotation on the part of the cup, but would fly up to the rim of the cup if the speed were ever so little increased, and would sink to the foot if the speed were ever so little reduced.

Fig. 129 shows a practical form of parabolic governor[1]. An

[1] From Mr J. Head's paper on "A Steam-engine Governor," *Proc. Inst. Mech. Eng.* 1871.

important feature is the air-cylinder at the top, forming a dash-pot, which is furnished with a small adjustable orifice through

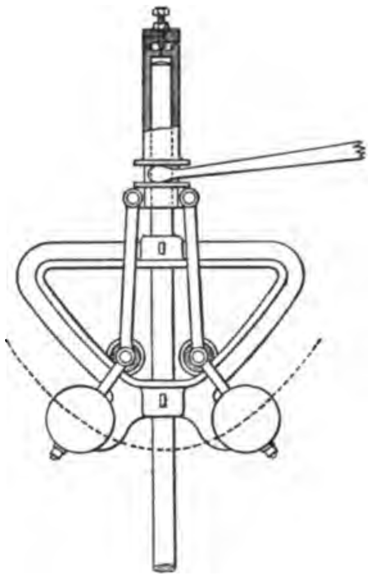


FIG. 129. Parabolic Governor.

which air is driven out or in as the balls rise or fall. The function of this is to check the tendency which the balls have to *hunt* (§ 190 below), or to fly violently in or out when the speed drops below or rises above the normal value.

185. Approximate Isochronism in Pendulum Governors. A useful approximation to the condition of isochronism can be reached in the conical pendulum governor by using crossed rods with the centres of suspension at a suitable distance from the axis. If each centre of suspension were so placed as to be at the centre of curvature of a parabolic arc which coincided, at the position corresponding to the normal speed, with the actual circular curve along which the balls rise and fall, the governor would be sensibly isochronous at that speed. By taking points a little nearer the axis for the two centres of suspension a margin of stability, always necessary in practice, is secured, but the governor is left nearly enough isochronous to be very sensitive. This crossed-rod type of governor, which is due to Farcot, is often

met with in a loaded form. An example is given in fig. 130. Loading a governor (whether the rods are crossed or open) need

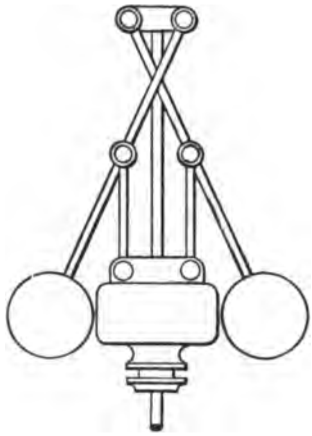


FIG. 130. Loaded Governor with Crossed Rods.

not affect the sensibility; it makes a higher speed necessary, but the proportion of the fluctuation of speed to the mean speed is not changed, provided the links are arranged in such a way that the vertical velocity-ratio of the load and the balls does not alter as the balls rise.

Another approximately isochronous form of gravity governor is Pröll's (fig. 131), which is interesting as exemplifying a different

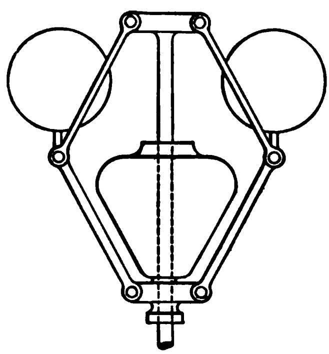


FIG. 131. Pröll's Governor.

method of reducing the stability of the pendulum type. Let the ball be supported not at the joint between the links as in the ordinary Porter governor but at the end of an arm projecting upwards and rigidly connected to the lower link. By a proper

choice of the length of this arm the controlling force may be made as nearly proportional to the radius as may be desired.

Pendulum governors of the stable class are occasionally loaded indirectly, the weight which forms the load being applied at some point in the lever by which the governor is connected with the valve. This allows the load and therefore the speed to be adjusted: further, by applying the load at the end of a cranked arm in the lever in such a way that it becomes less effective when the balls go out, the system can be made approximately isochronous.

186. Governors with spring control. Adjustment of sensitiveness. When springs furnish the controlling force, in whole or part, as in the governors shown in figs. 120 and 121, their tension is generally adjustable. This gives a convenient means of altering the speed; at the same time it affects the sensitiveness of the governor. In spring governors which are constructed so that the radial displacement of the balls produces a proportional change in the tension of the spring, the condition of isochronism can be approached, as nearly as may be wished, by giving the spring a suitable amount of initial tension. Thus in Mr Hartnell's apparatus, fig. 121, where the balls move in a nearly horizontal direction and gravity has almost nothing to do with the control, the governor can be made isochronous by screwing down the spring so that the initial force exerted by the spring (before the balls are displaced) is to the increase of this force by the displacement of the balls, as the initial radius of the balls' path is to the increase of that radius by the displacement. This makes F proportional to r, and therefore requires no change in n as the balls move out. Any greater initial tension would make the governor unstable, and a less tension is in fact necessary, in order that the sensitiveness may not be impracticably great.

187. Determination of the Controlling Force. Whatever be the method of control, by weights or springs or both, the controlling force F may generally be calculated for any assumed position of the balls. The simple pendulum governor both unloaded and loaded as in fig. 119 has already been considered. A case such as that of fig. 120 or fig. 121 presents no difficulty when the stiffness and initial tension of the spring are given. Slightly less simple cases of the loaded governor present themselves when the balls are not placed at the joints between the upper pair of links and

the lower pair which carry the load. Let the ball M (fig. 132) be fixed on the upper link AB, at any place either beyond B or between A and B. First find F_1, the stress in BC, from the consideration that BC and its twin link on the other side are in simple tension and support between them the load. Then the forces acting on ABM, namely F_1, the tension in BC, F_2 which is the weight of the ball, and F which is the force to be determined, are in equilibrium, and hence F is readily found by taking moments about A. We here treat F as the equilibrant instead of the resultant of the forces which are actually applied, for the sake of bringing the system into static equilibrium.

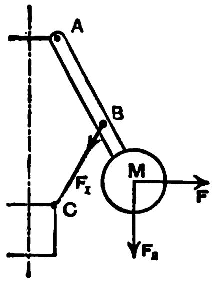


FIG. 132.

When the ball is carried by the link BC or by a piece rigidly connected to it as in Pröll's governor we may proceed thus (fig. 133):—The forces concerned in the equilibrium of the rigid piece CBM are (1) F_3 the half weight of the load acting at C, which is the vertical component of the pull at the joint C, (2) the horizontal component F_4 of the pull at the joint C, (3) the tension F_6 in the link AB, (4) the weight of M, or F_2, and finally (5) the force F which is to be determined. The resultant of F_3 and F_4 no longer acts along BC for there is a bending moment on the piece CBM. Compound F_2 and F_3 into a single force F_5. Since F_4 and F are horizontal, this vertical force F_5 must be wholly balanced by the vertical component of the stress in AB. Hence F_6 is found by drawing a right-angled triangle (fig. 134) with a line parallel to AB as hypotenuse and with a vertical side equal to F_5. Having found F_6 we are in a position to take moments about C in order to find F, which is now the only unknown force not acting through C.

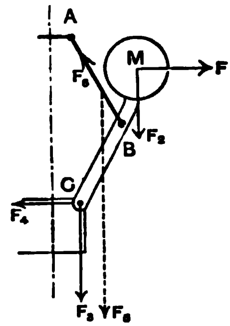


FIG. 133.

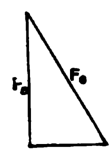


FIG. 134.

188. Influence of Friction. Power of the Governor. We may express the influence of friction on the behaviour of a governor by treating it as equivalent to a force with some

limiting value f, acting radially on each ball, in the same direction with the controlling force F when the balls are moving out and in the opposite direction when they are moving in. This makes the whole controlling force $F+f$ in the former case and $F-f$ in the latter. Let n be the speed proper to the force F alone, then if there were no friction any increase of speed above n would begin to alter the configuration, making the balls move out; but in consequence of friction this does not happen until the speed has increased by some finite amount Δn such that

$$n+\Delta n=\frac{1}{2\pi}\sqrt{\frac{F+f}{Mr}}.$$

Similarly, should the speed fall below the normal speed n proper to any configuration, friction prevents the balls from beginning to move in until the reduction of speed $\Delta' n$ is such that

$$n-\Delta' n=\frac{1}{2\pi}\sqrt{\frac{F-f}{Mr}}.$$

Hence in consequence of friction the speed may alter as much as Δn above and $\Delta' n$ below the normal speed n, while the position of the balls remains unchanged. From the above equations, if Δn be small relatively to n, as it always should be in practice, we have, approximately,

$$\frac{\Delta n}{n}=\frac{f}{2F}.$$

This variation of speed due to friction is independent of whatever further variation of speed the governor may allow in consequence of its equilibrium being stable (§ 183), and would of course be experienced even with a governor which except for friction was isochronous.

To keep the effects of friction within moderate limits it is essential that F should be great in comparison with f. The frictional resistance f proceeds partly from the joints of the governor itself but mainly from the throttle-valve spindle or from the expansion gear the position of which the governor has to regulate. A *powerful* governor, namely a governor with a large amount of controlling force F, is therefore required when any considerable amount of frictional resistance in the valve or gearing is to be overcome. With simple pendulum governors, the only way to secure power in this sense is to make the balls large.

Loaded governors have the advantage that great power may be secured with comparatively small revolving masses. The quality of powerfulness in a governor is increased whenever the controlling force is increased, whether by gravity loading or by the use of springs. From another point of view, the loaded governor (with the same revolving masses) is more powerful because it runs at a higher speed; but this is just because its controlling force F is greater.

189. Curves of Controlling Force. The consideration of sensitiveness and powerfulness in governors generally is greatly elucidated by using a graphic method, suggested by Mr Hartnell[1], of exhibiting the controlling force. Having found F for various positions of the balls, let a curve P_1P_2 (fig. 135) be drawn in

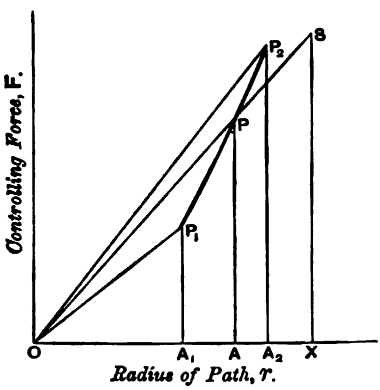


FIG. 135. Curve of Controlling Force.

which abscissæ represent r the radius of the balls' path and ordinates represent F. To find the configuration proper to any assigned speed n draw a line OS at such an inclination that $\tan SOX = 4\pi^2 n^2 M$, due regard being had to the scales of F and r. When the base OX is taken equal to unity on the scale used in plotting r, the value of SX is equal to $4\pi^2 n^2 M$ on the scale used in plotting F. Let P be the point in which this line cuts the curve of F. Then since

$$F = PA = OA \tan POA = 4\pi^2 n^2 r M,$$

it follows that the point of intersection P determines the radius OA at which the governor will run when the speed is n. Similarly the tangent of the angle which is made with the base by any other

[1] *Proc. Inst. Mech. Eng.* 1882.

line drawn from O to meet the curve, such as OP_1 or OP_2, is proportional to the square of the speed at the corresponding path-radius OA_1 or OA_2. Thus if OA_1 be set off to represent the least and OA_2 the greatest radius, corresponding to the positions giving full steam and no steam respectively, the inclinations of the lines OP_1 and OP_2 determine the whole range through which the speed will alter in consequence of the stability of the governor (apart from any effect of friction).

Further, if a pair of additional curves Q_1Q_2 and R_1R_2 be drawn as in fig. 136 to represent the values of $F+f$ and $F-f$ respectively, in relation to r, the diagram shows the additional changes of speed that are due to friction. The lowest possible speed is then determined by the inclination of the line OR_1, the highest by that of the line OQ_2.

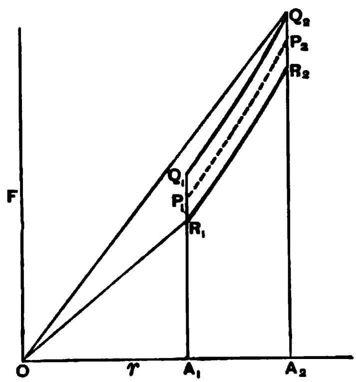


FIG. 136. Curves of Controlling Force, taking friction into account.

Again, the whole work done in altering the configuration of the governor, while the balls move out from A_1 to A_2 would be (for each ball) equal to the area $A_1P_1P_2A_2$ if there were no friction to be overcome: actually it is the area $A_1Q_1Q_2A_2$. And as the ball comes in from A_2 to A_1 the part of the stored energy which is recovered is measured by the area $A_2R_2R_1A_1$, the rest having been spent on friction. The area $P_1Q_1Q_2P_2$ is the work spent against friction while the governor is closing the throttle-valve or shifting the expansion gear from full steam to no steam.

The powerfulness of the governor is measured in a definite manner by the area $A_1P_1P_2A_2$, namely, the work stored and restored (save for friction) as the governor balls open or close throughout their range. In order that friction should cause no very serious irregularity in speed this area must be many times greater than the area $P_1Q_1Q_2P_2$ or $P_1P_2R_2R_1$. These last areas are equal if the friction f has the same value in closing as in opening the valve (as we have assumed above), but the construction shown in fig. 136 is evidently applicable whether f has or has not the same value during the rise and fall of the balls.

Again, the governor is stable provided the inclination of the curve to the axis OX be greater than the inclination of a line drawn from O to meet the curve at any point within the range of possible positions. Thus in fig. 135 the curve shows the governor to be stable because any line OP is less steep than the inclination of the curve itself at P. This is the condition of stability stated in § 180, namely, that the controlling force must increase more rapidly than the radius. A strictly isochronous governor would have for its curve of F and r a straight line passing, when produced, through O. If this condition were fulfilled by the line without friction P_1P_2, the line with friction Q_1Q_2, which lies above P_1P_2 at a more or less constant distance from it, would in general be less steep than a line from O drawn to meet it, which would mean that friction would make the otherwise neutral governor *unstable*. This is one reason why the isochronous governor is impracticable. The governor of fig. 136 is stable notwithstanding friction.

190. Hunting. Apart from the reason just stated it is indispensable to give a governor some margin of stability, especially when any change of speed takes some time to affect the supply of steam. An over-sensitive governor is liable to produce in the engine which it governs a state of forced oscillation called *hunting*. Several reasons contribute to produce this effect. When an alteration of speed begins to be felt, however readily the governor alters its form the engine's response is more or less delayed. The action of the regulator does not immediately take full effect upon the speed in consequence of the energy that is stored within the engine itself, not only in its moving parts but also in the steam that has passed the regulator and is still doing work in the engine. If the governor acts by closing a throttle-valve, the engine has still a capacious valve-chest on which to draw for steam. If it acts by changing the cut-off, its opportunity has passed if the cut-off has already occurred, and the control only begins in the next stroke. This lagging of effect is specially felt in compound engines, where that portion of the steam which is already in the engine continues to do its work for nearly a whole revolution after passing beyond the governor's control. The result of this storage of energy in an engine whose governor is too nearly isochronous is that whenever the demand for power suddenly falls

the speed rises so much as to force the governor into a position of over-control, such that the supply of steam is no longer adequate to meet even the reduced demand for power. Then the speed slackens, and the same kind of excessive regulation is repeated in the opposite direction. A state of forced oscillation is consequently set up. The tendency to hunt depends upon the fact that the rate at which steam does work is not immediately controlled when the load on the engine varies, but that there is a time-lag between any variation in the load and the proper corresponding variation in the action of the steam. A similar time-lag with a consequent tendency on the part of engine to hunt may proceed from another cause, which is independent of the storage of steam between the regulating valve and the engine piston. A sensitive governor, especially when it is of the relay type described below (§ 195), may take some time to come to its new position when the load is suddenly reduced. The governor begins to close the throttle-valve or to hasten the cut-off. But this takes time, and meanwhile the supply of steam is excessive and spends itself on the fly-wheel of the engine, giving it increased speed. By the time the supply is adjusted the speed has risen beyond its normal value, and a stage is reached when the regulating mechanism is carried too far and the supply is too much reduced. Thus a condition of forced oscillation may be set up even in cases where there is no storage of steam. The tendency is especially noticeable in engines with heavy fly-wheels running under light loads: under a heavier load the same engine may govern well without hunting. Again, hunting may be caused by the friction of the governor and of the regulating mechanism. Friction prevents the governor and regulator from beginning to change its position until the speed has changed by a finite amount, and when once the movement begins it goes beyond the point proper for steady control. The effect is aggravated by the momentum which the governor balls acquire in being displaced. Oscillations of the governor due to its own inertia are often prevented by introducing a *viscous* resistance to the displacement of the governor, which prevents the displacement from occurring too suddenly, without affecting the ultimate position of equilibrium. For this purpose many governors are furnished with a *dash-pot*, which is a hydraulic or pneumatic brake, consisting of a piston connected to the governor, working loosely in a cylinder

which is filled with oil or with air. An instance of the use of a dash-pot has already been mentioned in speaking of the parabolic governor of fig. 129.

191. Governor with horizontal axis. In some high-speed engines the governor balls or blocks instead of revolving about a vertical axis are arranged about the main shaft of the engine,

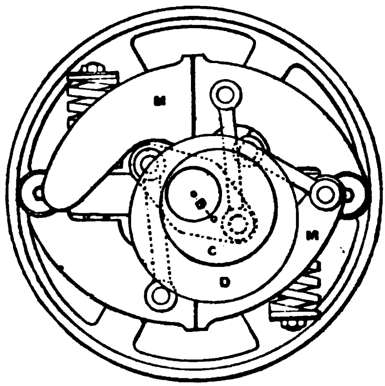


FIG. 137. Governor of Armington and Sims Engine.

sometimes within the fly-wheel, the control being given by springs. An example is shown in fig. 137, which is the governor of the Armington and Sims engine. Here the governor produces automatic variations of the cut-off by acting on the main slide-valve of the engine (there being no separate expansion-valve). The displacement of the revolving masses M, M changes both the throw and the angular advance of the eccentric, thereby effecting a change in the steam supply similar to that produced by "notching up" a link-motion. The eccentricity B is altered by the relative displacement of two parts C, D into which the eccentric sheave is divided. This relative displacement not only changes the length of B but gives it more or less of angular advance[1]. Governors resembling that of fig. 137 are often set on a horizontal axis in small high-speed engines.

192. Throttle-valve and automatic Expansion-gear. The throttle-valve, as introduced by Watt, was originally a disk turning on a transverse axis across the centre of the steam-pipe.

[1] For other governors of a like kind see Mr Hartnell's paper cited above.

It is now usually a double-beat valve (fig. 117) or a piston-valve. When regulation is effected by varying the cut-off, and an expansion-valve of the slide-valve type is used, the governor generally acts by changing the travel of the valve. Fig. 121 illustrates one usual mode of doing this, by giving the expansion valve its motion from an eccentric rod through a link the throw of which is varied by the displacement of the governor balls. In some forms of automatic expansion gear the governor acts upon the lap of the expansion valve. In others it acts by shifting the expansion eccentric round upon the shaft and so changing its angular advance. In others, again, it acts on an ordinary slide-valve through some form of link-motion or in such a way as has just been described.

193. Corliss and other Trip-gear. In large stationary engines the most usual plan of automatically regulating the expansion is to employ some form of trip-gear, the earliest type of which was introduced in 1849 by G. H. Corliss of Providence, U.S. In the Corliss system the valves which admit steam are distinct from the exhaust-valves. The latter are opened and closed by a reciprocating piece which takes its motion from an eccentric. The former are opened by a reciprocating piece, but are closed by springing back when released by a trip- or trigger-action. The trip occurs earlier or later in the piston's stroke according to the position of the governor. The admission-valve is opened by the reciprocating piece with equal rapidity whether the cut-off is going to be early or late. It remains wide open during the admission, and then, when the trip-action comes into play, it closes suddenly. The indicator diagram of a Corliss engine consequently has a nearly horizontal admission-line and a sharply defined cut-off. Generally the valves of Corliss engines are cylindrical plates turning in hollow cylindrical seats which extend across the width of the cylinder. Often, however, the admission-valves are of the disk or double-beat type, and spring into their seats when the trip-gear acts. Many forms of Corliss gear have been invented by Corliss himself and by others. One of these, the Spencer Inglis[1] trip-gear, by Messrs Hick, Hargreaves & Co., is shown in figs. 138 and 139. A wrist-plate A, which turns on a pin on the outside of the cylinder, receives a motion of oscillation from an eccentric. It opens the cylindrical rocking-valve B by

[1] *Proc. Inst. Mech. Eng.* 1868.

pulling the link *C*, which consists of two parts, connected to each other by a pair of spring clips *a*, *a*. Between the clips there is a

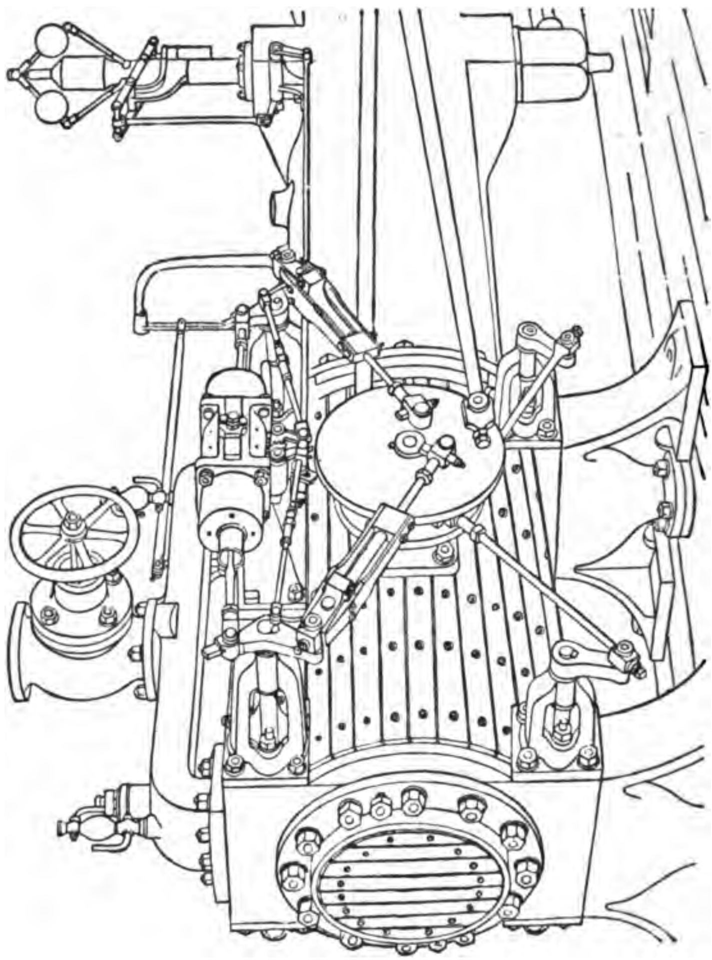


FIG. 138. Engine Cylinder with Corliss Valve-gear.

rocking-cam *b*, and as the link is pulled down this cam places itself more and more athwart the link, until at a certain point it forces the clips open. Then the upper part of the link springs back and allows the valve *B* to close by the action of a spring in the dash-pot *D*. When the wrist-plate makes its return stroke the clips re-engage the upper portion of the link *C*, and things are ready for the next stroke. The rocking-cam *b* has its position

controlled by the governor through the link E in such a way that when the speed of the engine increases it stands more athwart the link C, and therefore causes the clips to be released at an

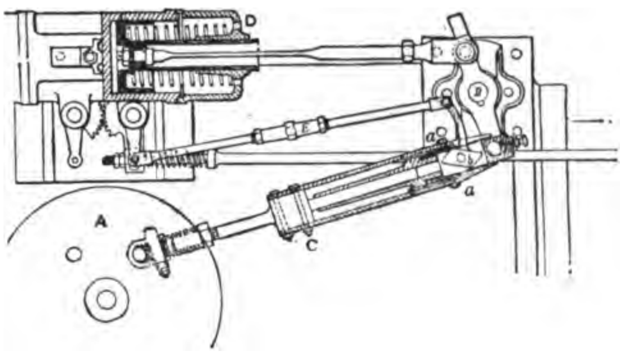


FIG. 139. Corliss Valve-gear, Spencer Inglis form.

earlier point in the stroke. A precisely similar arrangement governs the admission of steam to the other end of the cylinder. The exhaust-valves are situated on the bottom of the cylinder, at the ends, and take their motion from a separate wrist-plate which oscillates on the same pin with the plate A[1].

Fig. 140 shows a compact form of trip-gear by Dr Pröll. A rocking-lever ab is made to oscillate on a fixed pin through its centre by a connexion to the crosshead of the engine. When the end a rises, the bell-crank lever c engages the lever d, and when a is depressed the lever d is forced down and the valve e is opened to admit steam to one end of the cylinder. As a continues moving down a point is reached at which the edge of c slips past the edge of d, and the valve is then forced to its seat by a spring in the dash-pot f. This disengagement occurs early or late according to the position of the fulcrum piece g, on which the heel of the bell-crank c rests during the opening of the valve. The position of g is determined by the governor, which is of the kind already mentioned in § 185. A similar action, occurring at the other end of the rocking-bar ab, gives steam to the other end of the cylinder. In one form of Pröll's gear both ends of ab act on the same steam-valve, which is then a separate expansion-valve

[1] Numerous forms of Corliss gear are illustrated in W. H. Uhland's work on Corliss engines, translated by A. Tolhausen (London, 1879).

fixed on the back of a chest in which an ordinary slide-valve works.

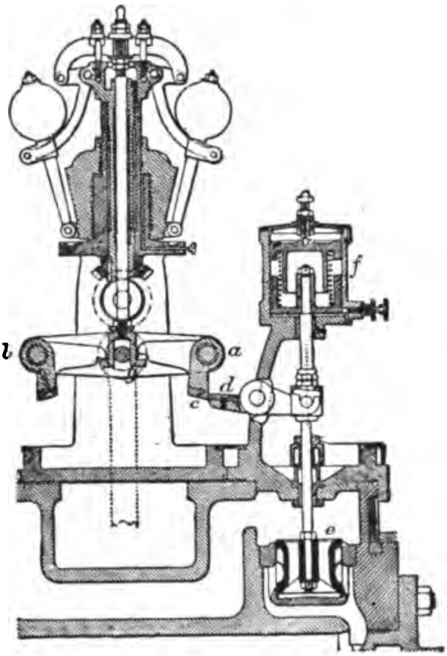


FIG. 140. Pröll's Automatic Expansion Gear.

194. Disengagement Governors. In the ordinary form of centrifugal governor the position of the throttle-valve, or the expansion-link, or the Corliss trigger depends on the configuration of the governor, and is definite for each position of the balls. In disengagement governors, of which the governor *A* shown on the right-hand side in fig. 141 is an example, any reduction of speed below a certain value sets the regulating mechanism in motion, and the adjustment continues until the speed has been restored. This is done by means of the wheel *c* which comes into gear with a wheel on the end of the spindle *a* when the speed falls below a certain limit. Similarly a rise of speed above a certain limit sets the regulating mechanism in motion in the other direction by putting *b* in gear with *a*. If the spindle *a* is connected to the regulator so as to give more steam when it turns one way and less when it turns the other, the speed at which the engine will run in equilibrium must lie between narrow limits, since at any speed

high enough to keep b in gear with a the supply of steam will go on being reduced, and at any speed low enough to bring c into

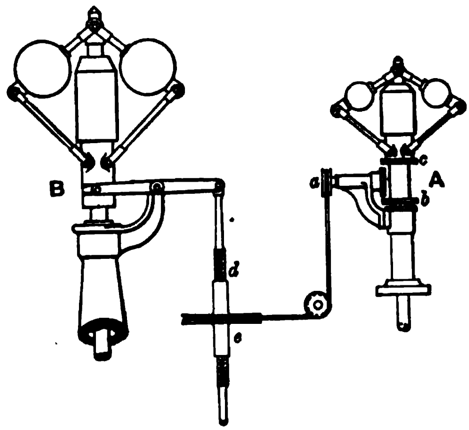


FIG. 141. Knowles's Supplementary Governor.

gear with a the supply will go on being increased. This mode of governing, besides being sensibly isochronous, has the important advantage that the power of the governor is not limited by the controlling force on the balls, since the governor acts by applying a portion of the power that is being developed by the engine to the work of moving the regulator. It is rarely applied to steam-engines, mainly because its action is too slow. This defect has been ingeniously remedied in the supplementary governor of Mr W. Knowles, who has combined a disengagement governor with one of the ordinary type in the manner shown in fig. 141[1]. Here the spindle a, driven by the supplementary or disengagement governor A, acts by lengthening the rod d which connects the ordinary governor B with the regulator. It does this by turning a coupling nut e which unites two parts of d, on which right- and left-handed screws are cut. Any sudden fluctuation in speed is immediately responded to by the ordinary governor. Any more or less permanent change of load or of steam-pressure gives the supplementary governor time to act. It goes on adjusting the supply until the normal speed is restored, thereby converting the control of the ordinary governor, which is stable, and therefore not

[1] *Proc. Inst. Mech. Eng.* 1884.

isochronous, into a control which is isochronous as regards all fluctuations of long period. The power of the combination, however, is limited to that of the ordinary governor *B*.

195. Relay Governors. Other governors which deserve to be classed as disengagement governors are those in which the displacement of the governor affects the regulator, not directly by a mechanical connexion, but by admitting steam or other fluid into what may be called a relay cylinder, whose piston acts on the regulator. In order that a governor of this class should work without causing the engine to hunt, the piston and valve of the relay cylinder should be connected by what is termed differential gear, the effect of which is that for each displacement of the valve by the governor the piston moves through a distance proportional to the displacement of the valve. An example of differential gear is shown in fig. 142. Suppose that the rod *a* is connected with the governor so that it is raised by an acceleration of the engine's speed. The rod *c* which leads from the relay piston *b* to the regulator serves as a fulcrum, and the valve-rod *d* is consequently raised. This admits steam to the upper side of the piston and depresses the piston, which pulls down *d* with it, since the end of *a* now serves as a fulcrum. Thus by the downward movement of the piston the valve is again restored to its middle position and the movement of the regulator then ceases until a new change of speed occurs. A somewhat similar differential contrivance is used in steam-steering engines to make the position of the rudder follow, step by step, every movement of the hand-wheel[1]; also, in the steam reversing gear which is applied to large marine engines, to make the position of the drag-link follow that of the hand-lever; and also in certain electrical governors[2]. The effect of adding a differential gear such as this to a relay governor or other disengagement governor is to convert it from the isochronous to the stable type.

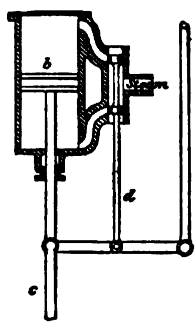


FIG. 142. Differential Gear for Relay Governor.

[1] See a paper by Mr J. MacFarlane Gray, *Proc. Inst. Mech. Eng.* 1867.

[2] Willans, *Min. Proc. Inst. C. E.*, Vol. LXXXI. p. 166.

196. Differential or dynamometric Governors. Another group of governors is best exemplified by the "differential" governor of the late Sir W. Siemens[1] (fig. 143). A spindle *a* driven by the engine drives a piece *b* (whose rotation is resisted by a friction-brake) through the dynamometer coupling *c*, consisting of a nest of bevel-wheels and a lever *d* which is loaded, the weight of the load acting at right angles to the plane of the paper. So long as the speed remains constant the rate at which work is done on the brake is constant and the lever *d* is steady. If the speed increases, more power has to be communicated to *b*, partly to overcome the inertia and partly to meet the increased resistance of the brake, and the lever *d* is displaced. The lever *d* works the throttle-valve or other regulator, either directly or by a steam relay. The governor is isochronous when the force employed to hold *d* in position does not vary; if the control of *d* is arranged so that the force tending to hold it in position increases when *d* is displaced, the governor is stable. A governor of this class may properly be called a dynamometric governor, since it regulates by endeavouring to keep constant the rate at which energy is transmitted to the piece *b*. In one form of Siemens's governor the friction-brake is replaced by a sort of centrifugal pump, consisting of a paraboloidal cup, open at the top and bottom, whose rotation causes a fluid to rise in it and escape over the rim when the speed is sufficiently great. Any increase in the cup's speed augments largely the power required to turn it, and consequently affects the position of the piece which corresponds to *d*.[2] Siemens's governor is not itself used to any important extent, but the principle it embodies finds application in a number of other forms.

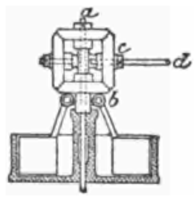


FIG. 143. Siemens's Governor.

The "velometer" or marine-engine regulator of Messrs Durham and Churchill[3] is a governor of the same type. In it the rotation of a piece corresponding to *b* is resisted by means of a fan revolving in a case containing a fluid, and the coupling piece which is the mechanical equivalent of *d* in fig. 143 acts on the

[1] *Proc. Inst. Mech. Eng.* 1853.
[2] *Proc. Inst. Mech. Eng.* 1866; or *Phil. Trans.* 1866.
[3] *Proc. Inst. Mech. Eng.* 1879.

throttle-valve, not directly but through a steam relay. In Silver's marine governor[1] the only friction-brake that is provided to resist the rotation of the piece which corresponds to *b* is a set of air-vanes. The inertia is, however, very great, and any acceleration of the engine's speed consequently displaces the dynamometer coupling, and so acts on the regulator in its effort to increase the speed of *b*.

Another example of the differential type is the Allen governor[2], which has a fan directly geared to the engine, revolving in a case containing a fluid. The case is also free to turn, except that it is held back by a weight or spring and is connected to the regulator. So long as the speed of the fan is constant, the moment required to keep the case from turning does not vary, and consequently the position of the regulator remains unchanged. When the fan turns faster the moment increases, and the case has to follow it (acting on the regulator) until the spring which holds the case from turning is sufficiently extended, or the weight raised. The term "dynamometric governor" is equally applicable to this form; the power required to drive the fan is regulated by an absorption-dynamometer in the case instead of by a transmission-dynamometer between the engine and the fan. In Napier's governor the case is fixed, and the reaction takes place between one turbine-fan which revolves with the engine and another close to it which is held from turning by a spring and is connected with the regulator.

197. Pump Governors. Pump governors form another group closely related to the differential or dynamometric type. An engine may have its speed regulated by working a small pump which supplies a chamber from which water or other fluid is allowed to escape by an orifice of constant size. When the engine quickens its speed the fluid is pumped in faster than it can escape, and the accumulation of the fluid in the chamber may be made to act on the regulator through a piston controlled by a spring or in other ways. This device has an obvious analogy to the cataract of the Cornish pumping-engine (§ 174), which has, however, the somewhat different purpose of introducing a regulated pause at the end of each stroke, or rather serves this purpose in addition to regulating the number of strokes per minute. The "differential

[1] *Brit. Ass. Rep.* 1859, p. 123. [2] *Proc. Inst. Mech. Eng.* 1893.

valve-gear" invented by Mr H. Davey, and successfully applied by him to modern pumping-engines, combines the functions of the Cornish cataract with that of a hydraulic governor for regulating the expansion[1]. In this gear, which is shown diagrammatically in fig. 144, the valve-rod of the engine (a) receives its motion from a

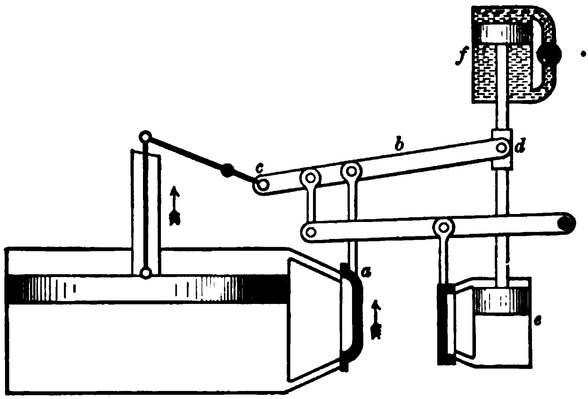


FIG. 144. Davey's Differential Valve-Gear.

lever b, one end of which (c) copies, on a reduced scale, the motion of the engine piston, while the other end (d), which forms the fulcrum, has its position regulated by attachment to a subsidiary piston-rod, which is driven by steam in a cylinder e, and is forced to travel at a nearly uniform rate by a cataract f. The point of cut-off is determined by the rate at which the main piston overtakes the cataract piston, and consequently comes early with light loads and late with heavy loads.

198. Governing Marine Engines. The government of marine engines is peculiarly difficult on account of the sudden and violent fluctuations of load to which they are subjected by the alternate uncovering and submersion of the screw in a heavy sea. However rapidly the governor responds to increase of speed by closing the throttle-valve, an excess of work is still done by the steam in the valve-chest and in the high-pressure cylinder. To check the racing which results from this, it has been proposed to supplement the control which the throttle-valve in the steam-pipe exercises by throttling the exhaust or by spoiling the

[1] *Proc. Inst. Mech. Eng.* 1874.

vacuum. With the same object Messrs Jenkins and Lee have given supplementary regulation by causing the governor to open a shunt-valve connecting the top with the bottom of the low-pressure cylinder, thus allowing a portion of the steam in it to pass the piston without doing work. In Dunlop's pneumatic governor[1] an attempt is made to anticipate the racing of the screw by causing the regulator to be acted on by the changes of pressure on a diaphragm which is connected by an air-pipe with an open vessel fixed under the stern of the ship. A plan has been used on small steamers by Mr W. B. Thompson to prevent the racing of the engines by working the valves from a lay shaft which is driven at a uniform speed by an entirely independent engine. So long as this lay shaft is not driven too fast the main engine is obliged to follow it; if the lay shaft is driven faster than the main engine can follow the main engine pauses so as to miss a stroke, and then goes on. Reversing the motion of the lay shaft reverses the main engine.

[1] *Proc. Inst. Mech. Eng.* 1879.

CHAPTER X.

THE WORK ON THE CRANK-SHAFT.

199. Fluctuations of Speed during any single revolution: function of the Fly-wheel. Besides those variations of speed which occur from stroke to stroke, which it is the business of the governor to check, there are variations within each single stroke over which the governor exercises no control. These are due to the varying rate at which work is done on the crank-shaft during its revolution. To keep them within reasonable limits is the function of the fly-wheel. It acts by forming a reservoir of energy to be drawn upon during those parts of the revolution in which the work done on the shaft is less than the work done by the shaft, and to take up the surplus in those parts of the revolution in which the work done on the shaft is greater than the work done by it. To accomplish this alternate storing and restoring of energy the fly-wheel has to undergo slight fluctuations of speed, whose range depends on the ratio which the alternate excess and defect of energy bears to the whole stock of energy the fly-wheel holds in virtue of its motion. The duty of the fly-wheel may be studied by drawing a *diagram of crank-effort*, which shows the work done on the crank in the same way that the indicator diagram shows the work done on the piston. The same diagram serves another useful purpose in determining the twisting and bending stress in the crank.

200. Diagram of Crank-effort. The diagram of crank-effort is best drawn by representing, in a curve drawn with rectangular co-ordinates, the relation between the torque or moment which the connecting-rod exerts to turn the crank and the angle

turned through by the crank. When the angle is expressed in circular measure, the area of the diagram is the work done on the crank. Or instead of selecting the turning moment and the angle turned through as the two co-ordinates, we may take the tangential effort on the crank-pin as one co-ordinate, namely the force which is found when the thrust against the crank-pin is resolved along the tangent to the crank-pin's path, the other component being directed towards the centre of the crank-shaft and consequently exerting no turning moment. The linear motion of the crank-pin in its circular path is then taken as the other co-ordinate of the crank-effort diagram ; and the area still represents the work done upon the crank.

Neglecting friction for the present, and supposing in the first place that the engine runs so slowly that the forces required for the acceleration of the moving masses are negligibly small, the moment of crank-effort is found by resolving the thrust P of the piston-rod into a component Q along the connecting-rod and a

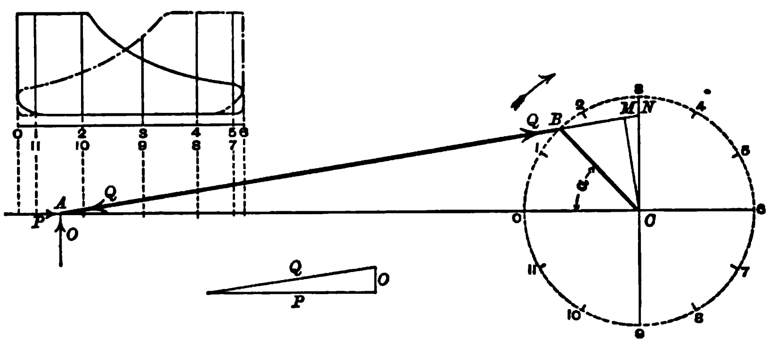


FIG. 145.

component O normal to the surface of the guide (fig. 145). The moment of crank-effort is

$$Q \cdot CM = P \cdot CN = Pr \sin \alpha \left(1 + \frac{r \cos \alpha}{\sqrt{l^2 - r^2 \sin^2 \alpha}}\right),$$

where CN is drawn perpendicular to the centre line or travel of the piston, r is the crank, l the connecting-rod, and α the angle ACB which the crank makes with the centre line. A graphic determination of CN is the most convenient in practice, unless the connecting-rod is so long that its obliquity is negligible, when the second term in the above expression vanishes. Fig. 146 shows the

diagram of crank-effort determined in this way for an engine whose connecting-rod is $3\frac{1}{2}$ times the length of its crank, and in

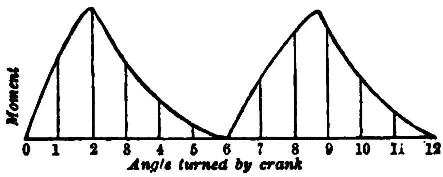


FIG. 146. Diagram of Crank-Effort.

which steam is cut off at about one-third of the stroke. The thrust P is determined from the indicator diagrams of fig. 145 by taking the excess of the forward pressure on one side of the piston over the back pressure on the other side, and multiplying this effective pressure by the area of the piston. The area of the diagram of crank-effort is the work done per revolution.

In the example for which this diagram is drawn it happens that there is very little compression of steam at the end of each back stroke, and consequently the forward pressure is greater than the back pressure throughout the whole of the stroke. In many cases, however, the back pressure rises so much toward the end of the stroke that the resultant thrust on the piston opposes its motion, the diagram of resultant steam pressure taking a form such as that sketched in fig. 147, and consequently the ordinates in the corresponding part of the crank-effort diagram become negative.

FIG. 147.

Another way of expressing the relation of the moment of crank-effort to the thrust P on the piston is to resolve the thrust Q along the rod into a component T in the direction of the tangent at B, and a component along BC. The former alone exerts a moment on the shaft, and its moment is $T \cdot CB$. To find T we have, by the principle of work, $T \cdot V_B = P \cdot V_O$, where V_B and V_O are the velocities of the crank-pin and piston respectively. Hence

$$T = \frac{P \cdot V_O}{V_B} = \frac{P \cdot IO}{IB},$$

where I is the instantaneous centre for the movement of the connecting-rod. I is found graphically by producing CB to meet a perpendicular to OC from O. Since $\frac{IO}{IB}=\frac{CN}{CB}$, the expression $T \cdot CB$ for the moment of crank-effort has the same value as that found before, namely $P \cdot CN$.

201. Effect of Friction. The friction of the piston in the cylinder and the piston-rod in the stuffing-box is easily allowed for, when its amount is known, by making a suitable deduction from P. Friction at the guides, at the cross-head, and at the crank-pin has the effect of making the stress at each of these places to be inclined to the rubbing surfaces at an angle ϕ, the angle of repose, whose tangent is the coefficient of friction. Hence the thrust O of the guide upon the cross-head instead of being normal to the surface of the guide, is inclined at the angle ϕ in the direction which resists the piston's motion (fig. 148); and the thrust along the connecting-rod, instead of passing through the centre of each pin, is displaced far enough to make an angle ϕ with the radius at the point where it meets the pin's surface. To determine this displacement of the line of thrust let a "friction-circle" be drawn about the centre of each pin, namely a circle with radius equal to $p \sin \phi$, where p is the actual radius of the pin. Any line drawn tangent to this circle will make the angle ϕ with the radius of the pin at the surface of the pin and will therefore satisfy the required condition as regards friction. The thrust of the connecting-rod must be tangent to both circles; it must

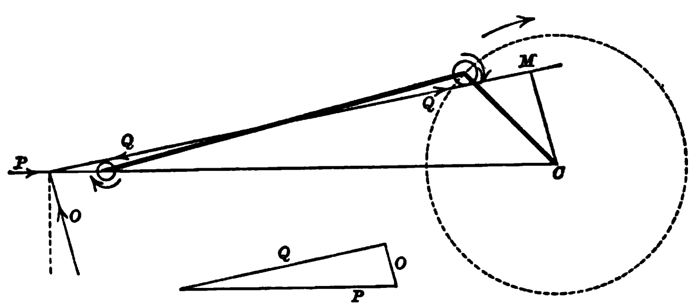


FIG. 148.

therefore be drawn as in fig. 140, so that it resists the rotation of the pins relatively to the rod. The direction of rotation of the

pins is shown by curved arrows in the figure, where the friction-circles are drawn to a greatly exaggerated scale. Finally, P (after allowing for the friction of piston-packing and stuffing-box) is resolved into O and Q, and then $Q \cdot CM$, the moment of Q on the shaft, is determined. This gives a diagram of crank-effort, correct so far as friction affects it, whose area is no longer equal to that of the indicator diagram. The difference, however, does not represent the whole work lost through friction in the mechanism, since the friction of the shaft itself, and of the valves and other parts of the engine which it drives, has still to be allowed for if the frictional efficiency of the engine as a whole is in question.

202. Effect of the inertia of the reciprocating pieces. The diagram of crank-effort is further modified when we take account of the inertia of the piston and connecting-rod, and the influence of inertia is generally much more important than that of friction. For the purpose of investigating the effects of the inertia of the reciprocating pieces, we may assume that the crank is revolving at a sensibly uniform rate of n turns per second. Let M be the mass of the piston, piston-rod, and cross-head in pounds, and a its acceleration at any instant in feet per second per second, the force required to accelerate it is $\frac{Ma}{g}$, in pounds-weight, and this is to be deducted in estimating the effective value of P. The effect is to reduce P during the first part of the stroke and to increase it towards the end, thereby compensating to some extent for the variation which P undergoes in consequence of any early cut-off. If the connecting-rod is so long that its obliquity may be neglected the piston has simple harmonic motion, and

$$a = -4\pi^2 n^2 r \cos \alpha,$$

when the crank has turned through any angle α from its dead point. More generally, whatever ratio the length l of the connecting-rod bears to that of the crank r,

$$a = -4\pi^2 n^2 r \left(\cos \alpha + \frac{rl^2 \cos 2\alpha + r^3 \sin^4 \alpha}{(l^2 - r^2 \sin^2 \alpha)^{\frac{3}{2}}}\right)^{*}.$$

* To prove this, let θ be the angle BAC of fig. 145; then

$$\theta = \sin^{-1}\left(\frac{r \sin \alpha}{l}\right).$$

The effect is to make, on the diagram of P, a correction of the character shown in fig. 149 where the broken line *cd* refers to the case of an indefinitely long connecting-rod and the full line *aeb* to the case of a connecting-rod $3\frac{1}{2}$ times the length of the crank. In a vertical engine the weight of the piston and piston-rod is to be added to or subtracted from P.

The form of the inertia line *aeb* of fig. 149 may be determined much more shortly and with sufficient accuracy for any graphic application by finding the points *a* and *e* and *b* as follows, and

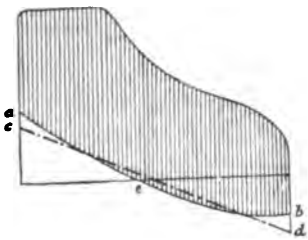


FIG. 149.

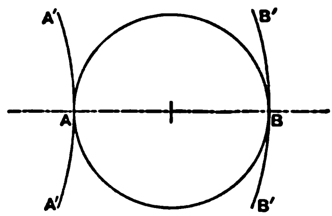


FIG. 150.

then sketching a smooth curve through these three points. The position of the point *e* in the stroke is found from the fact that since the acceleration is then zero the velocity of the piston is a maximum: this happens when the crank and connecting-rod are nearly at right angles (see next paragraph). The acceleration at

Hence, $$\frac{d\theta}{dt}=\frac{r\cos\alpha}{\sqrt{l^2-r^2\sin^2\alpha}}\frac{d\alpha}{dt}=\frac{2\pi nr\cos\alpha}{\sqrt{l^2-r^2\sin^2\alpha}}.$$

Differentiating again, and remembering that $\frac{d^2\alpha}{dt^2}=0$ since the rotation is assumed to be sensibly uniform, we obtain

$$\frac{d^2\theta}{dt^2}=\frac{-r(l^2-r^2)\sin\alpha}{(l^2-r^2\sin^2\alpha)^{\frac{3}{2}}}\left(\frac{d\alpha}{dt}\right)^2=\frac{-4\pi^2n^2r(l^2-r^2)\sin\alpha}{(l^2-r^2\sin^2\alpha)^{\frac{3}{2}}}.$$

Again, writing x for AC (fig. 145)

$$x=r\cos\alpha+l\cos\theta,$$

$$\frac{dx}{dt}=-r\sin\alpha\frac{d\alpha}{dt}-l\sin\theta\frac{d\theta}{dt},$$

and $$a=\frac{d^2x}{dt^2}=-r\cos\alpha\left(\frac{d\alpha}{dt}\right)^2-l\cos\theta\left(\frac{d\theta}{dt}\right)^2-l\sin\theta\frac{d^2\theta}{dt^2}.$$

Substituting the values found above for $\frac{d\alpha}{dt}$, $\frac{d\theta}{dt}$ and $\frac{d^2\theta}{dt^2}$, and putting $r\sin\alpha$ for $l\cos\theta$ and $\sqrt{l^2-r^2\sin^2\alpha}$ for $l\cos\theta$, this gives the expression in the text,

$$a=-4\pi^2n^2r\left(\cos\alpha+\frac{rl^2\cos 2\alpha+r^3\sin^4\alpha}{(l^2-r^2\sin^2\alpha)^{\frac{3}{2}}}\right).$$

a is the centrifugal acceleration due to the sum of the curvatures of the path of the crank-pin and of the arc AA' struck with l for radius (fig. 150). Similarly the acceleration at b (fig. 149) is due to the difference of these curvatures. Hence at a the acceleration is $\frac{v^2}{r}+\frac{v^2}{l}$ where v is the velocity of the crank-pin, and at b it is $\frac{v^2}{r}-\frac{v^2}{l}$. Substituting $2\pi nr$ for v in these expressions the acceleration of the piston is found to be

$$4\pi^2n^2r\left(1+\frac{r}{l}\right) \text{ and } 4\pi^2n^2r\left(1-\frac{r}{l}\right)$$

at a and b respectively.

203. Klein's construction for finding the acceleration of the Piston. This graphic method of finding the acceleration of the piston at any point in the stroke is shown in fig. 151. Produce AP to N and describe a circle with centre P and radius PN. Bisect the connecting-rod in E, and with E as centre and

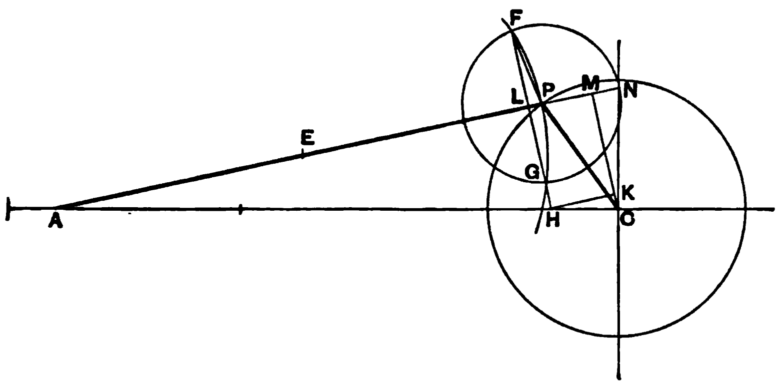


FIG. 151.

EP as radius draw a circular arc cutting the first circle in F and G. Join FG and produce it when necessary to cut the line AC in H. Then the length HC, when multiplied by the square of the angular velocity of the crank, gives the acceleration of the piston. In other words, if the length CP represents the radial acceleration of the crank-pin, HC represents the acceleration of the piston.

To prove this, draw CM perpendicular to PN and HK parallel to LM. Find also I the instantaneous centre of the connecting-

rod by producing CP through P to meet a line drawn at A perpendicular to AC. The triangle AIP (not shown in the diagram) is similar to NPC and $\frac{IP}{CP} = \frac{AP}{PN}$. For brevity we shall write ω for the angular velocity of the crank and ω' for the corresponding angular velocity of the connecting-rod. If v is the velocity of the crank-pin, $\omega = \frac{v}{CP}$ and $\omega' = \frac{v}{IP}$. Hence $\omega' = \omega . \frac{CP}{IP} = \omega . \frac{PN}{AP}$. The motion of the rod may be regarded as made up of (1) a translation with velocity v in the direction of the tangent to the crank-pin circle at P, and (2) an angular movement about P with angular velocity ω'. The acceleration of P along PC is $\omega^2 CP$. Resolve this into components along the rod and perpendicular to it. The component along the rod is $\omega^2 PM$, and this is also the acceleration of A in the direction AP, so far as A receives acceleration in consequence of the rod's movement of translation. The acceleration of A due to rotation of the rod about P is $\omega'^2 AP = \frac{\omega^2 . PN^2}{AP} = \frac{\omega^2 . PF^2}{AP} = \omega^2 LP$, since $\frac{LP}{PF} = \frac{PF}{AP}$. This acceleration is in the direction AP.

The total acceleration of A in the direction AP is therefore $\omega^2 (LP + PM) = \omega^2 LM$, the other component of acceleration being perpendicular to AP.

Hence a, the acceleration of A along AC, is

$$\omega^2 LM . \frac{HC}{HK} = \omega^2 HC,$$

which was to be proved.

Further, the component acceleration of A in the direction perpendicular to AP is $\omega^2 CK$. But this is made up of (1) a component of the general acceleration of the rod $\omega^2 CP$ due to its translation, namely $\omega^2 CM$, and (2) the acceleration due to rotation about P with angular velocity ω', namely $AP . \frac{d^2\theta}{dt^2}$.

Hence $$\omega^2 CK = \omega^2 CM - AP . \frac{d^2\theta}{dt^2}.$$

Or $$AP . \frac{d^2\theta}{dt^2} = \omega^2 LH^*.$$

* The author owes this extension to Mr S. Dunkerley. Another construction, by Rittershaus, for graphically determining the acceleration of the piston will be found in Unwin's *Elements of Machine Design*, Vol. II. p. 72.

Collecting the results we have:—

The angular velocity of the connecting-rod varies as PN, being equal to $\dfrac{\omega . PN}{AP}$.

The angular acceleration of the connecting-rod varies as LH, being equal to $\dfrac{\omega^2 LH}{AP}$.

The acceleration of the piston varies as CH, being equal to $\omega^2 CH$.

204. Position of the crank for which the piston has no acceleration. By means of Klein's construction, or otherwise, the position of the crank may be found for which the acceleration of the piston is zero and its velocity a maximum. This happens in the diagram (fig. 151) when H coincides with C. The corresponding crank-angle α is given by the cubic equation in $\sin^2 \alpha$,

$$\sin^6 \alpha - n^2 \sin^4 \alpha - n^4 \sin^2 \alpha + n^4 = 0,$$

where n is the ratio of the length of the connecting-rod to that of the crank[1]. The following table gives values of the angle for various values of n, and also values of the angle at which the connecting-rod is tangent to the crank-pin circle.

Ratio of connecting-rod to crank	Angle from the dead point at which the velocity of the piston is a maximum	Angle at which the connecting-rod is perpendicular to the crank
2	67° 42′	63° 26′
3	73° 11′	71° 34′
4	76° 43′	75° 58′
5	79° 7′	78° 41′
6	80° 48′	80° 32′
7	82° 2′	81° 52′
8	82° 59′	82° 52′
9	83° 44′	83° 40′
10	84° 20′	84° 17′

On comparing the two it will be seen that the rough approximation made by taking as the position of no acceleration the

[1] See Papers in *Min. Proc. Inst. C. E.* Vol. cxxiv. *et seq.* by Prof. Hill, Prof. Unwin, and Mr G. A. Burls. The above cubic equation in $\sin^2 \alpha$ for determining the position of maximum velocity will be found in Prof. Minchin's *Uniplanar Kinematics* (1882), p. 48.

place where the rod is tangent to the crank-pin circle introduces no serious error in ordinary cases.

205. Inertia of the Connecting-rod. The treatment of the inertia of the connecting-rod presents more difficulty than that of the piston. A rough approximation to the real effect is often arrived at by supposing part of the whole mass of the rod to be gathered at the cross-head, forming an addition to the mass which has simply a reciprocating motion, and the remainder to be gathered at the crank-pin, forming an addition to the rotating mass of the fly-wheel.

To obtain an exact solution the motion of the rod may be analysed as consisting of translation with the velocity of the cross-head, combined with rotation about the cross-head as centre. By means of this analysis, the force required for the acceleration of the rod is determined as the resultant of three components, namely, F_1, the force required for the linear acceleration a (which

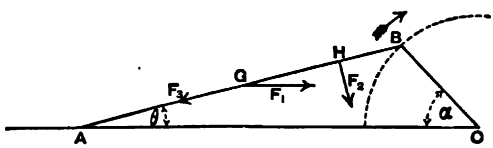


FIG. 152.

is the same as that of the piston); F_2, the force required to cause angular acceleration about the cross-head; and F_3, the force towards the centre of rotation, which depends on the angular velocity, and is equal and opposite to the so-called centrifugal force. Let θ as before be the angle BAC (fig. 152), so that $\frac{d\theta}{dt}$ is the angular velocity of the rod about A, and $\frac{d^2\theta}{dt^2}$ is its angular acceleration, and let M' be the mass of the rod. Then, using gravitational units,

$$F_1 = \frac{M'a}{g},$$

and acts through the centre of gravity G, parallel to AC;

$$F_2 = \frac{M'(AG)}{g} \frac{d^2\theta}{dt^2},$$

and acts at right angles to the rod through the centre of percussion H;

$$F_3 = \frac{M'(AG)}{g}\left(\frac{d\theta}{dt}\right)^2,$$

and acts along the rod towards A.

The values of a and of $\frac{d\theta}{dt}$ and $\frac{d^2\theta}{dt^2}$ in relation to the crank angle α have already been given, in the foot-note to § 202.

If now we imagine the directions of the forces F_1, F_2, F_3 to be reversed, these reversed forces will, when taken along with the weight of the rod, equilibrate the external forces applied to the rod at A and B. To draw the diagram of forces, refer to the joints A and B each of these reversed forces and also the weight. Then treat the rod as if it were a member in a frame, loaded at the joints and exerting simple thrust along its length. At A all the forces are known in direction but two of them are unknown in magnitude. These are found by drawing the polygon of forces for A, and then the polygon of forces for B gives the magnitude and direction of the force on the crank-pin.

206. Treatment of Inertia and Friction together. When in addition to the inertia of the rod, the friction at the cross-head and crank-pin is to be taken account of, the whole group of forces acting on the rod may be considered as follows. Compound forces equal and opposite to F_1, F_2, and F_3 into a single force R (fig. 153), which may be called the resultant resistance to

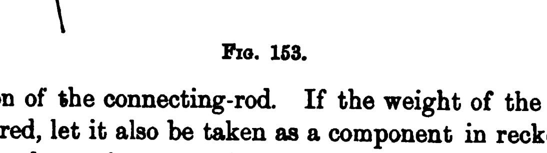


FIG. 153.

acceleration of the connecting-rod. If the weight of the rod is to be considered, let it also be taken as a component in reckoning R. Then the rod may in any position be regarded as in equilibrium under the action of the forces Q, R and S, where Q and S are the forces exerted on it by the cross-head and crank-pin respectively.

These three forces meet in a point p in the line of action of R, which point is to be found by trial, the condition being that in the diagram of forces, fig. 154, after the triangle POQ has been drawn, and the force R set out, the force-line S shall be parallel to a line drawn from p tangent to the friction-circle of the crank-pin, as shown in fig. 153. When this condition has been satisfied by trial, the value of S, which is the thrust on the crank-pin, is determined, and then $S.\overline{CM}$ is the moment of crank-effort. This method is due to Fleeming Jenkin, who applied it with great generality to the determination of the frictional efficiency of machinery in two important papers[1], the second of which deals in detail with the dynamics of the steam-engine. Fig. 155, taken from that paper, shows the diagram of

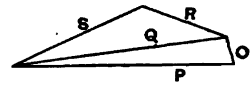


FIG. 154.

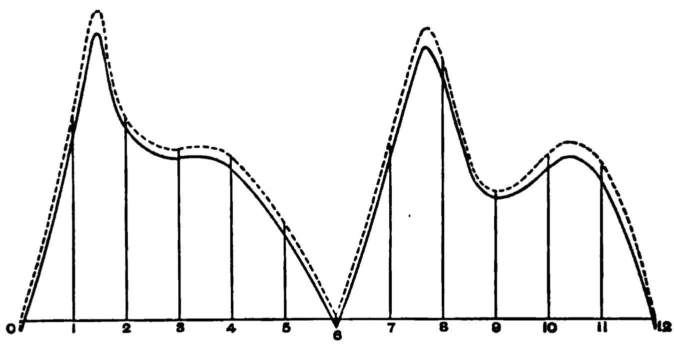


FIG. 155. Crank-Effort Diagram.

crank-effort in a horizontal direct-acting engine,—the full line with friction, and the dotted line without friction,—the inertia of the piston and connecting-rod being taken account of, as well as the weight of the latter. It exhibits well the influence which the inertia of the reciprocating parts exerts to equalize the crank-effort in the case of an early cut-off. The cut-off is supposed to occur pretty sharply at about one-sixth of the stroke. The engine considered is of practical proportions, and makes four turns per second; and the initial steam pressure is 50 lb. per square inch. It appears from the diagram that, with a slightly higher speed, or

[1] *Transactions of the Royal Society of Edinburgh*, Vol. XXVIII. p. 1 and p. 703.

with heavier rods, a better approach to uniformity in the crank-effort might be secured, especially as regards the stroke towards the crank, which comes first in the diagram; on the other hand, by unduly increasing the mass of the reciprocating pieces or their speed the inequality due to expansion would be over-corrected and a new inequality would come in.

In drawing crank-effort diagrams it is seldom necessary in practice to take account of the friction of the guide and of the pins, but the inertia of the piston, piston-rod and connecting-rod is of the utmost importance, especially in high-speed engines. The graphic method which is exhibited in figs. 153 and 154 of finding S, the thrust on the crank-pin, after R, the resistance to acceleration of the connecting-rod, has been determined, may of course be as readily applied when friction is neglected as when it is taken into account.

207. Forms of Crank-Effort Diagrams. When two or more cranks act on the same shaft the joint diagram, showing the resulting turning moment, is found by combining the separate diagrams of crank-effort for the several cranks. An example is shown in fig. 156, where the dotted lines are the separate diagrams

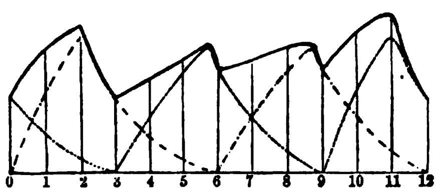


FIG. 156. Crank-Effort Diagram for Two Cranks.

for two cranks set at right angles to each other and the full line is the combined diagram. It is obvious that the inequalities of crank-effort are vastly reduced by using two cranks instead of one, and with three cranks the effort becomes still more uniform. An illustration of this is given in fig. 157, which also exemplifies the circular form in which the diagram of crank-effort is sometimes drawn. In this construction lines proportional to the moment are set off radially from a circular line which represents the zero of moment. The figure is one drawn by Kirk for a triple-expansion marine engine with three cranks at 120° from each other.

The curves show the resulting crank-effort, as determined from actual indicator diagrams and as affected by the inertia of the reciprocating parts. They are drawn for various numbers of revolutions per minute, which are indicated by the distinguishing

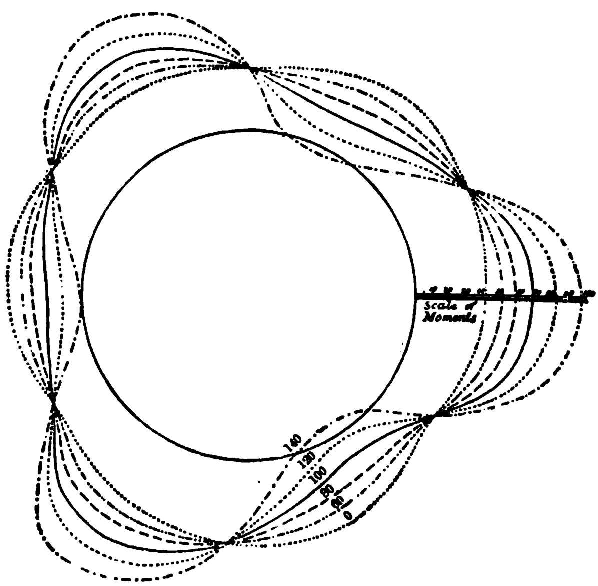


FIG. 157. Circular Diagram of Crank-Effort for a Three-Cylinder Engine.

numbers, the line marked 0 referring to an indefinitely slow motion.

As an opposite extreme to the nearly uniform crank-effort that is obtained by the use of three cranks the case may be named of an explosive gas- or oil-engine using the "Otto" cycle in which under the most favourable conditions the whole effective action on the crank takes place only in one single stroke out of two revolutions (or four strokes), two of the other three strokes being idle, and the third being that in which the explosive mixture is compressed before ignition (see Chapter XIV.). The student will find it an interesting exercise to draw a crank-effort diagram for such a case, extending the diagram over two revolutions to get a complete cyclic period, and then to apply the method described below of determining the size of fly-wheel which is necessary to prevent the speed from fluctuating beyond assigned limits. In the case of a gas-engine, however, it is not practically necessary to take into

account the inertia of the reciprocating parts in order to find the amount of energy that has to be alternately absorbed and given out by the fly-wheel. That is readily determined, from the indicator diagram, by comparing the work done on the piston during the single effective stroke with the mean amount of work done during the four strokes which make up the cycle. An example will be found in Chapter XIV.

208. Fluctuation of Speed in relation to the Energy of the Fly-wheel. The extent to which the fly-wheel has to act as a reservoir of energy is found by comparing the diagram of effort exerted on the crank-shaft by the piston or pistons with a similar diagram drawn to show the effort exerted by the crank-shaft throughout the revolution, in overcoming the resistance of the mechanism which it drives as well as the resistance due to its own friction. Like the driving effort, this resistance may be expressed as a torque or moment, or (dividing the moment by the radius of the crank) we may state the equivalent resistance referred to the crank-pin as a force acting always tangent to the crank-pin's path. In general, except in such cases as are offered by direct-acting pumping and blowing engines, or by engines which are compressing air or other gases, the resistance may be taken as having a constant moment on the

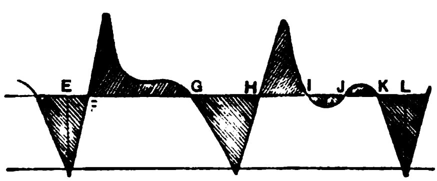


Fig. 158.

shaft, and the diagram of effort exerted by the crank-shaft is then a straight line, as *EFGHIJKL* in fig. 158. At *F*, *G*, *H*, *I*, *J*, and *K* the rate at which work is being done on and by the shaft is the same; hence at these points the fly-wheel is neither gaining nor losing speed. The shaded area above *FG* is an excess of work done on the crank, and raises the speed of the fly-wheel from a minimum at *F* to a maximum at *G*. From *G* to *H* the fly-wheel supplies the defect of energy shown by the shaded area below *GH*, which represents the amount by which the demand for work exceeds the supply; the speed of the

21—2

wheel again reaches a minimum at H, and again a maximum at I. The excesses and defects balance in each revolution if the engine is making a constant number of turns per second. In what follows it is assumed that they are only a small fraction of the whole energy stored up by the fly-wheel in virtue of its revolution, and consequently that the variations in speed are small in comparison with the mean speed. In practice the dimension and speed of the fly-wheel are chosen so that this is the case: indeed the chief object of the investigation is to find what amount of energy must be given to the wheel in order that the variations in speed may not exceed a prescribed range.

Let ΔE be the greatest single amount of energy that the fly-wheel has to give out or absorb, which is determined by measuring the shaded areas of the diagram and selecting the greatest of these areas; and let ω_1 and ω_2 be the maximum and minimum values of the wheel's angular velocity, which occur at the extremes of the period during which it is storing or supplying the energy ΔE. The mean angular velocity of the wheel ω_0 will be sensibly equal to $\frac{1}{2}(\omega_1 + \omega_2)$ if the range through which the speed varies is moderate. Let E_0 be the energy of the fly-wheel at this mean speed. Then

$$E_0 = \tfrac{1}{2} I \omega_0^2,$$

where I is the moment of inertia of the fly-wheel. Also

$$\Delta E = \frac{I(\omega_1^2 - \omega_2^2)}{2} = I\omega_0(\omega_1 - \omega_2) = 2E_0 \frac{(\omega_1 - \omega_2)}{\omega_0}.$$

The quantity $\frac{\omega_1 - \omega_2}{\omega_0}$, which we may write q, is the ratio of the extreme range of speed to the mean speed, and measures the degree of unsteadiness which the fly-wheel leaves uncorrected. If the problem be to design a fly-wheel which will keep q down to an assigned limit, the energy of the wheel must be such that

$$E_0 = \frac{\Delta E}{2q}.$$

The periodic fluctuations of speed which are due to the limited capacity the fly-wheel has for storing energy may be examined experimentally by means of the familiar chronographic device of causing a vibrator, such as a tuning-fork electrically maintained in vibration, to scribe its oscillations on a surface which moves with the fly-wheel shaft. A sheet of smoked paper clasping the shaft

itself forms a convenient surface, on which the fork draws an undulating line by means of a bristle or light pointed spring attached to one of its prongs. The fork should be mounted on a carrier such as the slide-rest of a lathe so that it may be kept moving slowly in a direction parallel to the axis of the shaft, in order that the records of successive revolutions may be traced on fresh portions of the smoked surface[1].

209. Reversal of thrust at the joints. Prevention of reversal of the thrust in single-acting engines. Let the diagram of resultant steam thrust upon the piston be represented by the line SS as in fig. 159 for the two successive strokes of a revolution, the line being drawn in such a way as to show that the steam is pushing the piston towards the crank when it lies above the base, and is pulling the piston away from the crank

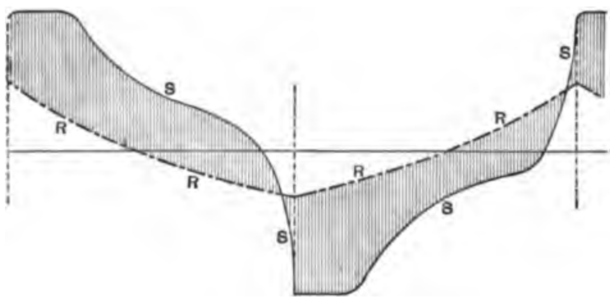


FIG. 159.

when it lies below the base. Let the line RR represent in the same way the forces that are used up in producing the acceleration of the reciprocating pieces. Then the points at which the steam curve SS crosses the inertia curve RR mark the places at which the direction of thrust at the bearings becomes reversed. If in drawing RR the mass of the piston, piston-rod and cross-head only is taken account of the intersection of the two curves will show at what places the thrust changes its sign at the cross-head pin. But if the mass of the connecting-rod also has been added in calculating the forces represented by this curve, the points where SS crosses RR will relate to the reversal of thrust on the bearing surfaces of the crank-pin. Two inertia lines may

[1] For examples of the use of this method of finding q see Mr H. B. Ransome's paper, *Min. Proc. Inst. C. E.* Vol. XCVIII., or the *Society of Arts Report on Trials of Motors for Electric Lighting* (1889).

be drawn, one referring to the masses between the steam and the crosshead pin, the other to the whole reciprocating mass, up to the crank-pin. Since the bearings are necessarily somewhat loose to admit of lubrication and free turning of the pins in their brasses, a sudden reversal of the thrust from pull to push at either joint will give rise to a knock. To prevent an engine from knocking badly the clearance at the bearings is of course to be kept as small as possible, and the form of the thrust diagram (fig. 159) has to be such that when the steam and inertia curves cross each other the change from positive to negative in the distances intercepted between them shall be gradual.

In some forms of high-speed single-acting engines this change is entirely avoided, and in that case the bearings may be left slack. In the Willans engine for example the back is the active end and the piston and connecting-rod are kept in compression throughout the revolution. During the stroke towards the crank this is their natural state, except when the speed is so great as to make the point *a* of fig. 149 rise above the steam thrust line. But during the out-stroke there is nothing happening in the cylinder, except a little compression towards the end of the stroke, to provide the force that is required to reduce the velocity of the reciprocating pieces after the point of maximum velocity (near mid-stroke) has been passed. Hence unless special provision for this force were made the connecting-rod would be pulling instead of pushing the crank-pin during the later portion of the out-stroke. In the Willans engine the special provision consists in an air-cylinder, the piston of which is arranged tandem with the steam piston (or steam pistons, in the case of a tandem compound engine of this single-acting class). The air in this cylinder begins to be compressed early in the out-stroke and becomes more and more compressed to the end, the energy which is expended in compressing it being given out again during the in-stroke or effective stroke of the engine. The force exerted by the compressed air (along with that exerted by the steam during the up-stroke) is arranged to be always in excess of the force that is required for the (negative) acceleration of the pistons and rods, and hence the thrust both at the crosshead and at the crank-pin is continuously a push, never a pull[1].

[1] See Discussion on High-Speed Motors, *Min. Proc. Inst. C. E.* Vol. LXXXIII., 1885.

An example will help to make this clear. Let *dd*, fig. 160, be the indicator diagram of a single-acting vertical engine to

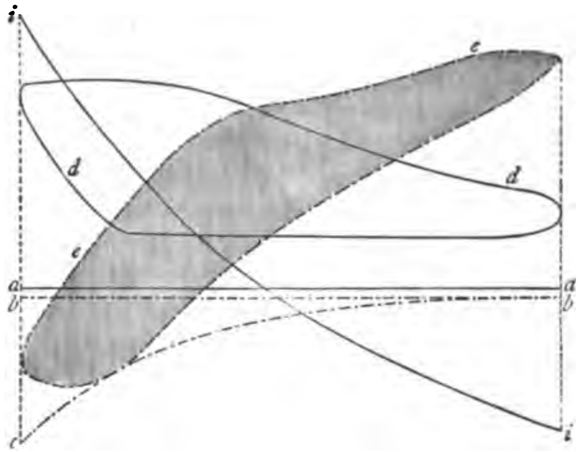


FIG. 160.

which steam is admitted on the top of the piston only. It is required to find what amount of air-compression on the part of an air-buffer piston will serve to keep the thrust on the crank-pin from changing its sign at any point in the revolution. The line *aa* is the atmospheric datum line, representing the constant pressure which acts below the piston of the steam cylinder. The line *ii* represents the forces due to the inertia of the whole reciprocating mass which is carried by the crank-pin—namely, the piston and piston-rod of the steam cylinder and of the air-buffer and also the connecting-rod. The forces due to inertia are represented per square inch of piston area, to the same scale as the steam pressures. Let the diagram *ee* be drawn to compound the forces due to inertia with those due to steam pressure. In other words, let its ordinates above or below the datum line *aa* be the excess of the ordinates of *dd* above those of *ii*. So long as the figure *ee* lies above the datum line *aa*, the steam pressure pushing the piston down exceeds the force necessary for acceleration, and consequently there is push, not pull, at the crank-pin. But when the figure *ee* comes below the line *aa* the force required for acceleration exceeds the force exerted by the steam. We must however take account of the weight of the reciprocating pieces, which assists the steam pressure. This is readily done by shifting

the datum line down to *bb*, the distance *ab* representing the weight, expressed in lbs. per square inch of piston area. The forces which are left to be balanced by the compression of air in the air-buffer are represented by the projection of *ee* below *bb*. Any such compression line for the air cylinder as *cc*, touching or lying wholly below the projecting part of *ee*, will therefore serve to prevent the force at the crank-pin from ever changing from a push into a pull. In practice, in the Willans engine, there are generally two, or three, steam pistons arranged tandem on each piston-rod, and the steam diagram to be used in the foregoing construction would be a diagram representing the sum of the pressures on the two, or three, pistons.

Again, in Messrs Mather and Platt's form of high-speed single-acting vertical engine steam is admitted to the under side of each piston and the rods are kept in tension instead of compression. In this case the connecting-rod should always pull against the crank-pin, and never push. This is effected by adding a balance-piston at the top of the cylinder which is continuously exposed to the full pressure of the steam on its lower side. The case is illustrated in fig. 161. There *dd* is the indicator diagram and

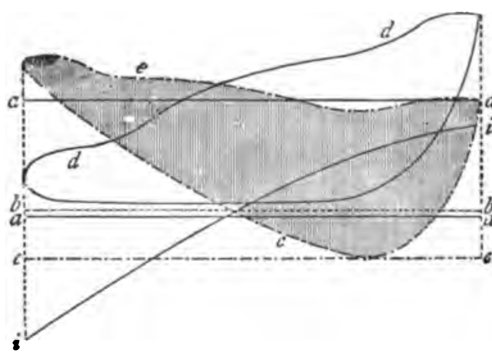


FIG. 161.

ii the inertia line as before. The datum line *aa* is not the atmospheric line *a'a'* but a line showing the pressure in the space above the piston, which space is connected with the condenser through the exhaust pipe. As before, the indicator diagram *dd* is compounded with the inertia line to give the figure *ee*: the datum line is shifted, in this case, *up* to *bb*, to allow for the weight of the reciprocating parts, which now hinders instead of helping

the steam to keep the rod in tension. Then the projection of *ee* below *bb* shows what has to be provided for on the part of the steam balance piston, and the pressure which the steam exerts on it must be at least equal to the height *bc* in order to prevent a reversal of force at the crank-pin.

210. Balancing. An important matter in the kinetics of the steam-engine is the balance of its working parts. A machine is said to be perfectly balanced when the relative movements of its parts have no tendency to make it vibrate as a whole. In other words, perfect balance implies that the reactions of those forces that are required for the acceleration of the parts should neutralize each other in every phase of the motion, so that no resultant reaction is ever felt by the bed-plate of the machine. A perfectly balanced machine would be self-contained as regards the stresses between the parts and would run steadily without foundations. Actual machines rarely do more than approximate to this condition.

In steam-engines and other machines using a piston, connecting-rod and crank, an approximate balance can be attained, so far as forces parallel to the direction of the stroke are concerned, by connecting to the crank-shaft two or more masses which revolve with it and are arranged so that the radial forces required to accelerate them are together equal and opposite to the force required to accelerate the piston, piston-rod, connecting-rod and crank-pin when the piston is at its dead-point. A single revolving mass is insufficient to effect this balancing, for it cannot be placed just opposite the crank-pin, and if placed alongside it still leaves an unbalanced couple the moment of which tends to rock the bed-plate about an axis perpendicular to the stroke and to the

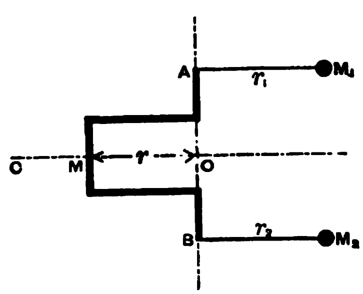


FIG. 162.

axis of the shaft. By using a pair of masses this is avoided. In the figure (fig. 162) AB is the axis of the shaft and CO is the direction of the stroke. The reciprocating pieces are treated as a single mass M concentrated at the crank-pin, the effective length of the crank to the centre of the pin being r. Let balancing masses M_1 and M_2 be set opposite the crank with their centres of gravity at distances r_1 and r_2 respectively from the axis of the shaft. Then to avoid having any resultant centrifugal force parallel to CO the condition must hold that

$$M_1\omega^2 r_1 + M_2\omega^2 r_2 = M\omega^2 r,$$

whence

$$M_1 r_1 + M_2 r_2 = Mr.$$

And, similarly, to avoid any centrifugal couple tending to twist the machine about an axis perpendicular to the plane of the figure,

$$M_1 r_1 \overline{OA} = M_2 r_2 \overline{OB}.$$

In an engine with a single crank the balance masses M_1 and M_2 are generally made equal and placed symmetrically on the two sides of the crank.

A balance arrived at in this way is not perfect, even as regards forces parallel to the direction of the stroke. The assumption that the whole reciprocating mass M might be treated as if it were collected at the crank-pin is more and more wide of the truth the shorter the connecting-rod is. With a short rod there is, as we have seen above, an important difference at the two dead-points in the values of the force necessary for accelerating the reciprocating parts. Hence at one dead-point (namely when the piston is nearest to the crank) the forces due to the balance-weights are in excess, and at the other dead-point they are in defect, of the force that has to be balanced. The treatment of the whole reciprocating mass as if it were collected at M is equivalent to ignoring the shortness of the connecting-rod.

With this reservation a balance may be secured in respect of forces acting in the plane of the sketch (fig. 162), namely the plane containing the line of stroke and the axis of the shaft, and all that has been said above relates to forces in that plane only. As regards forces at right angles to that plane the piston and piston-rod require no balancing for they suffer no acceleration at right angles to the plane in question, and only a part of the connecting-rod can be taken as approximately sharing the crank-pin's motion in this respect. Hence the balancing masses which

have been calculated for the forces in the plane COA will be altogether excessive in respect of forces in the direction normal to that and will give rise to vibrations in other directions. The conditions that are necessary to secure a balance in the two planes are incompatible, and the best results will in general be arrived at by a compromise.

In machines that can be anchored down to a massive foundation a state of defective balance only results in straining the parts and causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is of much more consequence in locomotive engines, where any bad want of balance produces oscillations that might be dangerous. Even in stationary engines, however, the question of balance is sometimes particularly important, as for instance in high-speed engines for the electric lighting of towns, which are often placed at stations where any vibration of the ground, communicated to neighbouring buildings, would be a serious nuisance.

In locomotives the existence of two cranks adds a slight complication to the problem of determining proper balance-weights to avoid horizontal oscillations; and this of course applies generally to engines with more than one crank. Let M, M' be the masses of the reciprocating parts referred to the crank-pins. Suppose that the balancing masses are to be carried (as is usual) on the driving wheels A and B. To balance M alone would require two masses, namely M_1 on A and M_2 on B, placed opposite to M and satisfying the conditions that

$$M_1r_1 + M_2r_2 = Mr,$$

and $$M_1r_1\,\overline{AP} = M_2r_2\,\overline{BP}.$$

Similarly to balance M' alone would require two masses, M_1' on A and M_2' on B, placed opposite to M' and satisfying two corresponding conditions.

For the two balancing masses on each wheel there may then be substituted a single mass on each wheel occupying a position

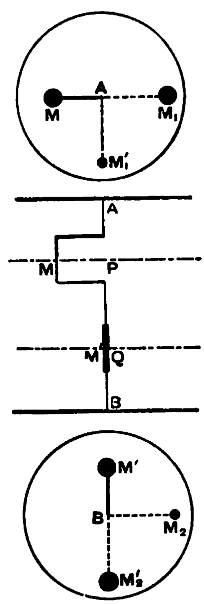


FIG. 163.

between the two, but nearer to M_1 on wheel A and nearer to M_2 on wheel B.

To find the amount and position of a single mass M_0 which, at any radius r_0, may be substituted for the masses M_1 and M_1' in the same plane, we have to make the force $M_0\omega^2 r_0$ form the resultant of the two forces $M_1\omega^2 r_1$ and $M_1'\omega^2 r_1$. Let ϕ be the angle made by the radius at which M_0 is to be placed with the radius AM_1.

Then $$M_0^2 r_0^2 = M_1^2 r_1^2 + M_1'^2 r_1^2,$$

and $$\tan\phi = \frac{M_1'}{M_1}.$$

In general r_0 may be made equal to r_1, and in that case

$$M_0 = \sqrt{M_1^2 + M_1'^2}.$$

Taking the case of an engine which is symmetrical about a longitudinal centre line midway between the cranks, write $2D$ for the distance AB and $2d$ for the distance PQ, and let $r_1 = r_2 = r_0$. Then $M_1 = M_2'$ and $M_2 = M_1'$, and the conditions become,

$$M_1 + M_2 = M\,.\frac{r}{r_0},$$

$$M_1(D-d) = M_2(D+d),$$

from which

$$M_1 = M\,.\frac{r}{r_0}\,.\frac{D+d}{2D},$$

and $$M_2 = M\,.\frac{r}{r_0}\,.\frac{D-d}{2D}.$$

The single mass M_0 which is to be placed at the radius r_0 as a substitute for M_1 and M_2 on each wheel will then be

$$\sqrt{M_1^2 + M_2^2} = M\,.\frac{r}{r_0}\,.\frac{\sqrt{D^2+d^2}}{D\sqrt{2}},$$

and its position is defined by making ϕ such that

$$\tan\phi = \frac{D-d}{D+d}.$$

211. Balance of the longitudinal forces in High-speed Engines. By the longitudinal forces those forces are to be understood which are due to the reciprocating masses and act parallel to the piston-stroke. Thus in a high-speed vertical engine the forces in question are those which act vertically. If it

were not for the obliquity of the connecting-rod these forces, for each piston, would have the same magnitude at the top of the stroke as at the bottom, and hence if there were two cranks 180° apart, with equally heavy pistons on both, the frame of the engine would at every instant be suffering equal upward and downward thrusts. With a single pair of cranks these equal and opposite thrusts would form a couple, causing the engine to rock about a transverse axis, but there would be no rising or falling of the engine as a whole. This however would only be true if the connecting-rods were indefinitely long. In a real engine with two opposed cranks the shortness of the rods disturbs this equality between the upward and downward thrusts, and gives rise to a thumping of the engine on its base. This is illustrated by the upper diagram in fig. 164, which shows by the line *AA* the forces due to the inertia of one piston and by the line *BB* the forces due to the inertia of another piston side by side with the

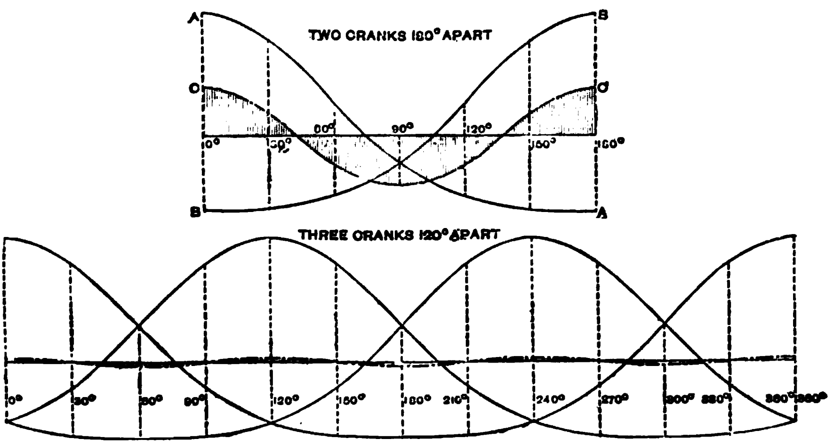


FIG. 164.

first and worked off a crank 180° from its crank, each connecting-rod being four times the length of the crank. The forces are shown in relation to the angle turned through by the crank, the base of the diagram corresponding to half a revolution. The line *CC* which is drawn by compounding the two shows how an up-and-down thrust results, with a period twice that of the revolution of the engine. The diagram shows how this thrust is produced as a

consequence of the shortness of the connecting-rods causing unequal forces at the two ends of the stroke.

On the other hand, it is interesting to notice that with three cranks, set at 120° from each other, the resultant up-and-down thrust almost vanishes. This is well shown by the lower diagram in fig. 164. The three cranks are assumed to carry equal reciprocating masses. This arrangement practically does away with any hammering of the engine as a whole upon its bed, but leaves a certain amount of rocking about a transverse horizontal axis, in consequence of the three forces not being in the same plane. But this rocking may be got rid of by using six cranks on the same shaft, in two symmetrical sets of three each, grouped so that the rocking couple produced by one set balances that produced by the other. With this arrangement a practically perfect balance is obtained[1].

[1] See a paper by Mr M. Robinson on Single-Acting High-Speed Engines, *Jour. Inst. Elect. Engineers*, Vol. xxiv. p. 434 (1895). For the curves of fig. 164 the author is indebted to Mr Dunkerley. Reference should be made to Professor Dalby's treatise on the Balancing of Engines (London, E. Arnold, 1902) for a general discussion of the subject.

CHAPTER XI.

THE PRODUCTION OF STEAM.—BOILERS.

212. Heating Surface, in Boiler and Feed-water Heater. In the transfer of energy from fuel to steam two stages may be distinguished. First, the potential energy of combustion is transformed into actual heat, which shows itself in the raised temperature of the furnace gases; and, second, the heat of the furnace gases passes by conduction through the *heating surface* into the water of the boiler. The furnace gases serve as a vehicle for the conveyance of heat from the furnace or fire-box where it is generated to the various parts of the heating surface, some of which may be a long way from the actual seat of combustion. The heating surface is made up of the surface of the furnace or combustion-chamber, so far as that is brought into contact with the water, and of the flues or tubes through which the hot gases pass on their way to the chimney. The effectiveness of any portion of the heating surface depends mainly on the difference in temperature between the gases on one side and the water on the other, and on the freedom with which steam, when formed, can escape from the surface. Differences in specific conductivity and in thickness of metal affect the result less than might be expected, partly on account of the resistance which is offered to the passage of heat through the scale which forms on the metallic surface, but mainly because, on the one side, the steam that is generated on the surface itself opposes the conduction of heat and must give place to unevaporated water before much more heat can be taken in, and, on the other side, the gas which has parted with its heat to the metal must escape and be replaced by hot gas in order that the transfer of heat may continue. It is the circulation

of the substances on either side, more than the conductivity of the plate itself, that determines the rate at which heat will pass through a boiler shell.

As the gases traverse the flues or tubes their temperature falls, until they finally escape at a temperature which is necessarily somewhat higher than that of the water to which they have been yielding up their heat. This temperature, however, is not necessarily higher than or even as high as the temperature of the steam, for after ceasing to be in contact with the boiler proper the gases may continue to give up heat to a *feed-water heater*, which is a set of pipes through which the comparatively cold feed-water passes on its way to the boiler. The feed-water heater virtually forms an extension of the heating surface, with the advantage that it is more effective for the transfer of heat than an equivalent extension of the boiler surface proper would be, on account of the lower temperature of the contents; and it allows the initial temperature of the feed-water, instead of the temperature of the steam, to form the lower limit to which the temperature of the gases might conceivably be allowed to fall. Conduction however would become so slow if the temperature of the gases approached this limit that in practice they are always considerably hotter. Even after passing a feed-water heater, the gases rarely have a temperature less than 400° Fah. When the draught through the fire is maintained by means of a chimney there is this independent reason for allowing the gases to escape at a relatively high temperature that the draught depends on the contents of the chimney being lighter than the air outside, and this lightness is secured by their being considerably hotter than the atmospheric air.

213. Draught. The furnace gases are made up of the products of combustion along with a quantity of air of dilution which passes through the furnace without undergoing chemical change. For the complete combustion of each pound of coal about 12 pounds of air are required to furnish the necessary oxygen, and usually about 12 pounds more have to enter as air of dilution. The greater part of this air enters through the grate, between the fire-bars on which the burning fuel rests, but some air has to be admitted above the fire to complete the burning of the combustible gases. This is specially necessary when fresh

coal is thrown on the fire and volatile hydrocarbons are being given off. The furnace door has apertures to allow a small part of the air to pass through it, and these are often made adjustable in area.

A *natural* or *chimney* draught is one which is produced wholly by the lightness of the contents of the chimney. A *forced* draught is one in which other means are taken to produce a difference between the pressure of the air inside and outside of the furnace. A fan, for instance, may be used to force the draught, either by extracting air from the flues or by blowing air into a closed room from which the furnace takes its supply. Or a jet of steam may be made to blow in the chimney, producing a partial vacuum there on the principle of the jet pump.

With a forced draught it is easy to produce much more difference in pressure above and below the grate than can readily be produced by means of a chimney, and consequently to compel the entrance of a larger quantity of air through the fuel, with the result that a much larger quantity of coal can be burned per square foot of grate. A furnace using chimney draught does not as a rule burn more than 20 lbs. of coal per hour per square foot of grate, but with forced draught the combustion may go on at four or five times this rate and still be fairly perfect.

Further, when the draught is forced the combustion is intensified and localised, and it is found that a smaller proportion of air will suffice for dilution. Instead of the 24 lbs. or so of air which chimney draught requires per lb. of coal, 18 lbs. or less will serve. Hence with a forced draught the temperature of the furnace gases is higher, and consequently the effectiveness of the heating surface is increased. Again, since the proportion of air passing through the furnace is reduced by forcing the draught, the proportion of heat lost in the hot gases is also reduced, provided the heating surface be extended sufficiently to make them leave the flues at no higher temperature than before.

But the theoretical advantage of forced draught in respect of efficiency does not stop here. When the draught does not depend on the action of a chimney there is no need to let the escaping gases have any higher temperature than is imposed by the condition, already indicated, that they must be reasonably hotter than the temperature of the feed. With a chimney, on the other hand, as much heat is necessarily wasted as will keep the

temperature of the escaping gases up to the comparatively high value necessary to maintain the draught. A chimney being an exceedingly inefficient form of heat-engine, the heat which is expended in maintaining its draught is vastly greater than the equivalent of the work that a fan would do in producing the same draught, or even than the heat that would have to be supplied to an engine employed in driving the fan.

In practical instances in which the draught is forced, namely, in locomotives and in some marine and a few land engines, the theoretical advantages of forced draught, in respect of efficiency, are imperfectly realised. The draught has generally been forced with the object of increasing the power of a given boiler rather than of securing a high efficiency. The motive has been to burn more coal per square foot of grate surface, and to get a higher temperature in the furnace gases, so that more water may be evaporated in a boiler of given weight. This is incompatible with high efficiency, for to secure efficiency the heating surface must be largely increased in order that it may deal with the augmented total quantity and higher temperature of the furnace gases. It is clear however that the most efficient boiler would be one using a strong mechanically forced draught, with a relatively small area of grate and a relatively very large heating surface, extended by the use of a feed-water heater, so that not only the gases should be cooled as far as possible before escaping, but that the proportion of air to coal should be as small as is consistent with thorough combustion.

214. Sources of loss of Heat. Ordinarily about seven-tenths and rarely more than eight-tenths of the potential energy of the fuel are conveyed to the steam. The remaining two or three-tenths are accounted for as follows:—(1) waste of fuel in the solid state by dropping through the grate; (2) waste of fuel in the gaseous and smoky state by imperfect combustion; (3) waste of heat by external radiation and conduction; and (4) waste of heat in the escaping gases due mainly to their high temperature, but partly also to their containing as one of the products of combustion a certain amount of steam-gas which passes off uncondensed. Of these sources of waste the first is generally trifling and the fourth is the most important. If we assume the air of dilution to be 12 lbs. the whole quantity of gas escaping from the chimney is

25 lbs. per lb. of coal burnt. The specific heat of this gas is nearly that of air, say 0·24 thermal units (§ 36). Hence about 6 thermal units are lost, per lb. of fuel burnt, for every degree by which the temperature of the escaping gas is allowed to exceed the lowest attainable limit. A chimney temperature of 600° Fah. is not unusual, and if we take 100° Fah. as a limit determined by the temperature of the feed-water, this represents a more or less preventable waste of 3000 thermal units, or in round numbers one-fifth of the whole energy of the coal.

215. Chimney Draught. In a chimney draught the "head" (usually stated in inches of water pressure) under which the current of air is kept up is equal to the amount by which the weight of a column of air in the chimney falls short of the weight of a corresponding column of outside air. Except for their excess of temperature the contents of the chimney would be heavier than the air outside nearly in the ratio of $n+1$ to n, where n is the number of pounds of air which have taken up 1 lb. of fuel in passing through the furnace. The actual density of the gases is less than that of the air outside in the proportion $\frac{\tau}{\tau_0}\left(\frac{n+1}{n}\right)$ to 1, where τ and τ_0 are the absolute temperatures inside and outside respectively. The difference in actual density multiplied by the height of the chimney gives the effective head. This head is used up partly in setting the column of air in motion and partly in overcoming the resistance to its passage which is offered by the flue, by the chimney itself, and by the grate. With a forced draught and a short chimney the resistance of the grate is the most important of these items; with a tall chimney on the other hand the resistance of the chimney itself comes to be so considerable that an increase of height produces almost no increase of draught, and may even diminish the draught if the sectional area is at all reduced in the added part. Under such conditions also there is a limit in the extent to which the draught will be assisted by letting the chimney temperature remain high. In raising the temperature of the chimney gases a stage is reached at which the gain in head and consequently in velocity of current is more than counterbalanced by the diminution of density, and if the gases are hotter than this the amount of gas passing through the chimney in a given time is actually reduced. In cases where the resistance

is practically all met with after the gases have become heated—in other words, when the resistance of the grate is a very small part of the whole, the maximum draught is produced when the contents of the chimney have a density equal to half that of the air outside[1]. Assuming 24 lbs. of air to be admitted per lb. of fuel this condition is reached when the temperature in the chimney is about 600° Fah. or about the melting point of lead. When the resistance of the grate is a substantial part of the whole a rather higher temperature will make the draught a maximum. No advantage whatever is gained by making the temperature higher than corresponds to maximum draught, and on the score of thermal efficiency a lower temperature is to be preferred, as diminishing the heat lost in the escaping gases.

216. Boilers for Stationary Engines. Cornish and Lancashire Types. Most modern boilers are internally fired, that is to say, the furnaces are more or less completely enclosed within the boiler. Externally fired boilers (except when they are of the water-tube type to be subsequently described) are in general distinctly less efficient than internally fired boilers; they are, however, used to some extent at coal-pits and other places where fuel is specially cheap or where the waste heat of other furnaces is to be utilized. Their usual form is that of a horizontal cylinder with convex ends; the strength both of the main shell and the ends is derived from their curvature, and no staying is required. Generally the heating surface is entirely external and is of very limited extent.

In large stationary boilers the forms known as the "Cornish" and "Lancashire" are the most common. The shell of these boilers is a long horizontal cylinder with flat ends, and within this, stretching from end to end within the water space, is a single large tube in the Cornish form and two parallel tubes in the Lancashire form, each tube containing a furnace at one end and communicating at the other end with external flues which are arranged to make nearly all the external surface of the shell below the water line act as part of the heating surface. The remainder of the heating surface is given by the large tube or tubes which contain the furnace, with the addition generally of several short cross tubes containing water, which traverse the main furnace tube

[1] See Rankine's *Steam-Engine*, p. 289.

at right angles to its length and not only serve the purpose of enlarging the heating surface, but promote circulation in the water, and strengthen the main tube. Fig. 165 shows a Cornish boiler in longitudinal section, and Fig. 166 is a cross section which shows the arrangement of the external flues. The furnace extends from the front up to the bridge of fire-brick *C*. In continuing their passage beyond this through the main tube or flue the hot gases come in contact with the cross-tubes, or Galloway tubes, *DD*, which have a somewhat conical form so that they may allow the steam formed in them to rise readily. At the end of the internal flue the gases are diverted downwards into the external flue *B*, and having traversed it towards the front of the boiler they are made to rise into the two side flues *AA*, by which they again pass to the back end and thence to the chimney. The form of the Lancashire boiler is essentially the same, except that there are two furnace tubes placed side by side, the diameter of the shell being larger. Fig. 167 is the cross-section of a Lancashire boiler. In a modified form of this boiler, introduced by Mr Galloway, the two furnace tubes unite beyond the bridge into one with a flat section, which is prevented from collapsing by having a number of Galloway tubes in it to act as stays.

The shell of a Lancashire boiler is commonly about 28 feet long, with a diameter of 7 feet, which allows each of the two furnace tubes to be 2 feet 9 inches wide. A boiler of this size, burning 20 lbs. or so of coal per hour per square foot of grate, will evaporate about 6000 lbs. of water per hour, or enough to yield, with an efficient condensing engine, from 300 to 400 indicated horse-power[1].

In boilers of this type the curvature of the cylindrical shell and furnace tubes enables them to resist the pressure of the steam: only the flat ends require to be stayed. This is done by means of gusset-stays *EE* (fig. 165), which tie the end plate to the circumference of the shell, and often also by means of longitudinal stay-bolts stretching from end to end within the water space. The furnace flues are made up of a series of short welded lengths united by joints which give the whole tube stiffness to resist collapse, but leave it some freedom to bend when the top expands more than the bottom through the unequal action of the

[1] For particulars of the Lancashire boiler see a paper by Mr L. E. Fletcher, *Proc. Inst. Mech. Eng.* 1876.

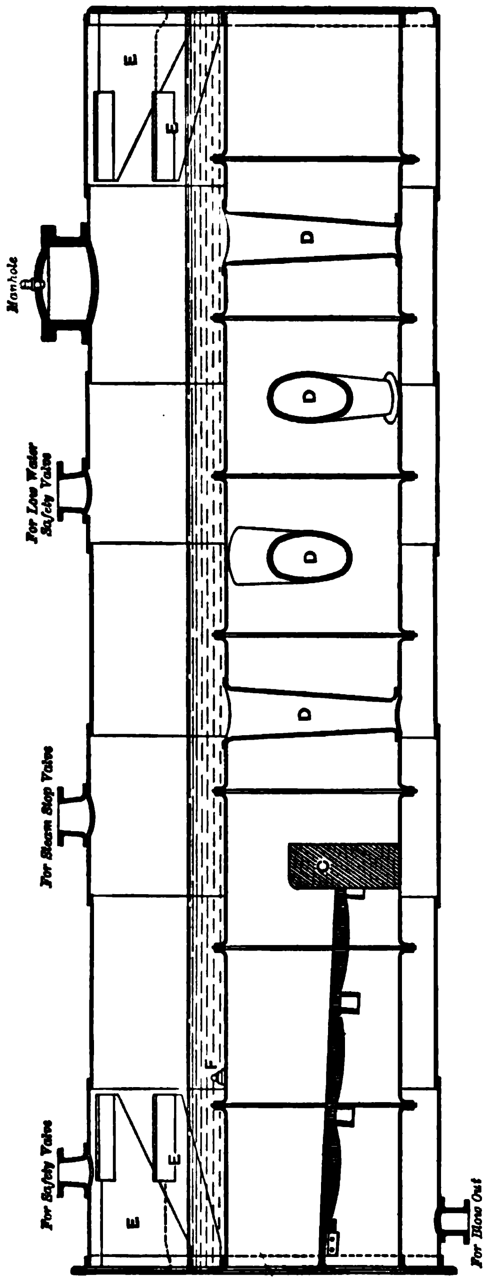


FIG. 165. Longitudinal section of Cornish Boiler. *C* bridge of fire-brick forming termination of the grate. *DD* Galloway tubes to stiffen the flue tube, to promote circulation of the water, and to increase the heating surface. *EE* gusset stays to support the flat ends. *F* fusible plug.

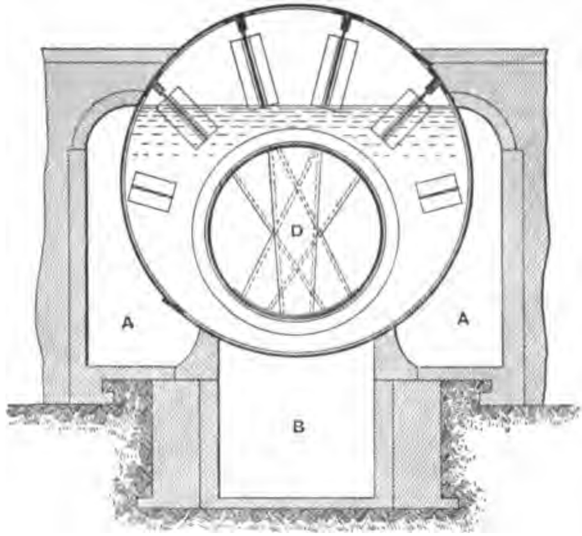


FIG. 166. Transverse section of Cornish Boiler.

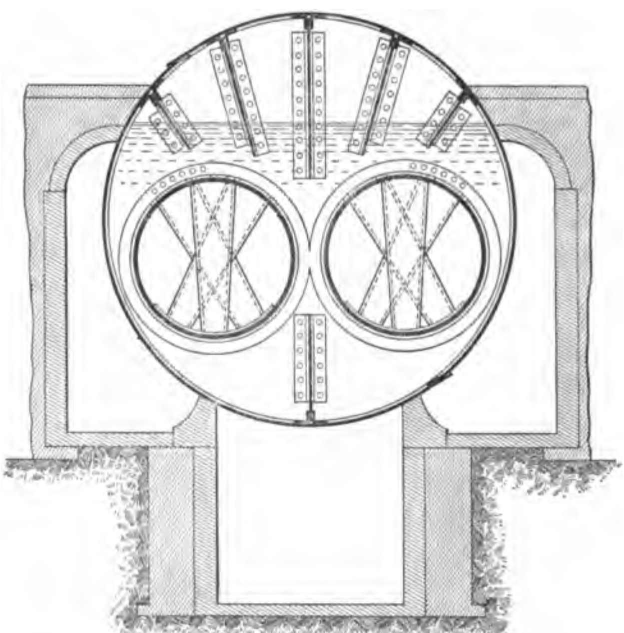


FIG. 167. Transverse section of Lancashire Boiler.

fire. To provide for unequal expansion is one of the most important points in the design of a boiler: when it is neglected a racking action occurs which induces leakage at the joints and tends to tear the plates. For this reason the furnace flues are attached only to the end plates and not to the cylindrical part of the shell, and the stays of the end plates are arranged to leave these plates some freedom to bulge out and in when the flues lengthen and contract.

217. Boiler Mountings. The steam dome, which used to be an ordinary feature in boilers of this type, is now generally omitted, and steam is taken direct from the steam space within the shell through a perforated "antipriming" pipe, from which it passes through a nozzle on the top of the boiler (fig. 165) to the stop-valve. The other openings on the top of the shell are the man-hole, on which a cover is bolted, and two openings for safety-valves. One of these valves is frequently of the dead-weight type, in which the force by which the valve is held closed is furnished by the direct action of a pile of weights: in many cases however springs and weighted levers are used. The second safety-valve is often arranged to form what is called a low-water safety-valve, being connected to a float in such a way that the valve will open if the water is allowed to sink below a safe level. At the bottom of the shell there is another nozzle for the blow-out cock, and in the front plate, below the furnace tubes there is another man-hole. Feed water is supplied by a pipe which enters through the front plate on one side near the top of the water and extends a good way in, distributing the water by holes throughout its length. A pipe at the same level on the other side serves to collect scum. On the top of each furnace is a fusible plug (*F*, fig. 165) which melts if the furnace crown become overheated. On the front plate are a pair of glass gauge tubes showing the level of the water within and a pressure gauge of the Bourdon type. This important fitting consists of a metal tube, oval in section, which is bent into a nearly circular form. One end is closed and is free to move: the other is held fixed and is open to the steam. The pressure of the steam tends to make the oval section rounder, and consequently tends, through 'anticlastic' bending, to straighten the tube. The free end accordingly moves through a small distance which is proportional to the excess of pressure within

the tube above the atmospheric pressure to which its outer surface is exposed, and this movement is magnified by an index turning on a dial. Most of the fittings which have been mentioned are common to boilers of all types.

218. Multitubular Boilers. In several other forms of boiler an extensive heating surface is obtained by the use of a large number of small tubes through which the hot gases pass. This construction is followed in locomotive and marine boilers, and boilers of the typical locomotive and marine forms (to be presently described) are, especially the former, frequently used with stationary engines. The multitubular construction is also applied in some instances to boilers of the ordinary cylindrical form by making a host of small tubes take the place of that part of the flue or flues which lies behind the bridge, or by using small tubes as channels through which the gases return from the back to the front after they have passed through the main flue. Still another form of tubular boiler is an externally fired horizontal cylinder filled with return tubes extending from back to front. In all these forms the tubes are placed within the water space of the boiler. Except in locomotives the tubes are commonly of iron, and a usual diameter is about 3 inches. They give so much heating surface that the outside surface of the shell need not be used, and hence in a tubular boiler the external flues are dispensed with which are a necessary feature of the Cornish or Lancashire type.

219. Vertical Boilers. In the boilers which have been referred to the axis of the cylindrical shell is horizontal. But the cylinder may be turned up on end and the boiler take a vertical form, the grate of course remaining horizontal and forming the floor of a fire-box to which access is given by a door on the side of the cylindrical shell. Large vertical boilers are now uncommon, but the type is a very usual one for boilers of small power. It has the drawback that the free surface of the water from which the steam rises is comparatively small and consequently the steam rises with a higher velocity, which increases the risk of priming. Fig. 168 shows an ordinary small vertical boiler with Galloway tubes across the upper part of the fire-box; and Fig. 169 is another form, in which the water tubes are curved channels which allow the water to circulate from the space round the sides of the fire-box to the space above the crown. In other forms of vertical

boiler the heating surface is increased by water tubes which hang from the crown of the fire-box and are closed at the lower end,

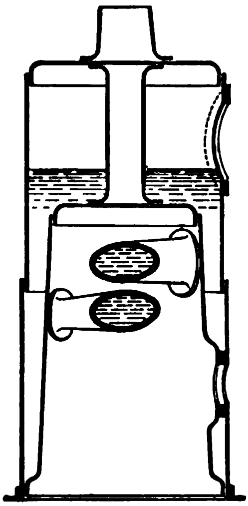


FIG. 168.

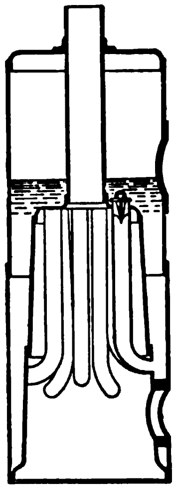


FIG. 169.

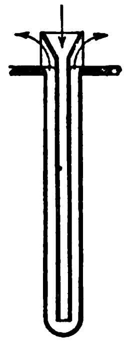


FIG. 170.

circulation of water being maintained in them by means of a partition in the form of an inner tube inside of which water flows down to allow an upward movement of water and steam to be maintained between the inner tube and the outer. Tubes of this kind are called Field tubes, and are particularly used in the boilers of fire-engines and in other cases where steam has to be got up with the least possible delay. A section of a Field tube is given in fig. 170, with arrows to indicate the manner in which the water circulates.

220. Water-tube Boilers. Many forms of boiler have been designed in which the firing is external, and the heating surface is made up of the outer surface of numerous tubes or other small sectional parts, through which a circulation of water is kept up in virtue of the differences in density between the hotter and colder portions of the water. In ordinary boilers the circulation is more or less casual: when a bubble of steam is detached from any part of the heating surface its place is taken by water which may come in from any side. In a properly designed water-tube boiler the circulation is systematic: water enters each of the tubes at one end and passes through in a continuous thin stream, becoming

partly converted into steam as it goes. The tubes generally deliver into a separating vessel, from the upper part of which the steam-pipe takes its supply, while water collects in the lower part to be returned by gravity to the lower end of the tubes. Boilers of this type can be constructed so as to have, with their contents, a relatively small total weight in proportion to the rate at which they can make steam, which is a distinct merit in respect of marine and especially of naval use. For erection in some situations such as basement rooms they have the advantage that they can be brought together in small pieces. Further they are easily made strong enough to resist exceptionally high pressures owing to the absence of any large shell: an early tubular boiler, for instance, designed by Mr Loftus Perkins delivered steam at 500 lbs. per square inch.

A successful example of this type is the Babcock and Wilcox boiler (fig. 171), the heating surface of which is almost wholly

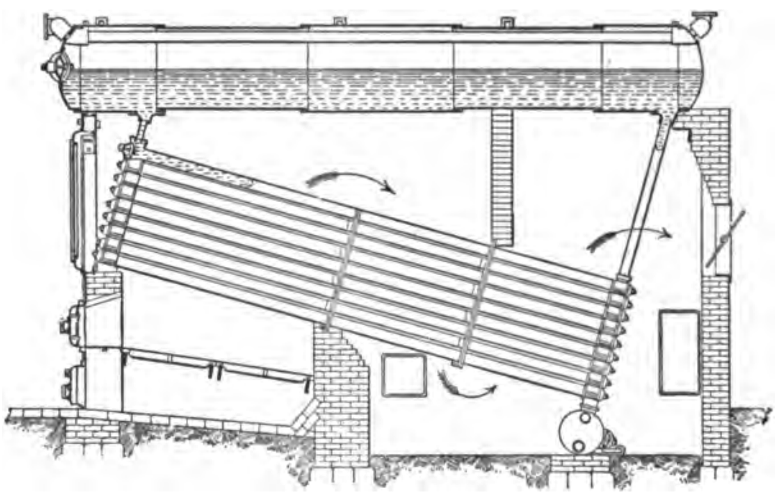


FIG. 171. Babcock and Wilcox Boiler.

composed of a series of inclined tubes up which water circulates in parallel streams. These are joined at their ends by cast-iron connecting boxes to one another and also to a horizontal drum on the top in which the mixture of steam and water which rises from the tube undergoes separation. At the lowest point of the boiler is another drum for the collection of sediment. The route taken

by the hot gases is indicated by arrows in the figure. Root's boiler is another of very similar form. In the Belleville boiler, now extensively adopted in the British and other navies, the tubes are grouped in sets, each set forming a flattened helix through the whole length of which the water rises from the sediment chamber to the separating drum. The tubes are of steel, about 4½ inches in diameter. They slope up with a gradient of about 1 in 25, alternately to left and right, forming a zigzag of straight lengths, which are made continuous by malleable cast-iron junction boxes. Each set or section has in all a length of about 150 feet through which the current of steam and water passes from end to end. Eight or more such sections stand side by side to make up the complete boiler. A non-return check-valve at the bottom assists in preventing the flow from taking place in the wrong direction. When the boiler begins to make steam the circulation occurs in a series of gusts, the check-valve closing while each gust makes its way up through the zig-zag of tubes. Harrison's boiler is a group of small globular vessels of cast-iron strung like beads on rods which tie them together. The Herreshof boiler is a continuous coil of tube, arranged as a dome over the fire. Feed water is pumped slowly through the coil, and turns to steam before it reaches the end. Here the circulation is mechanically forced instead of being due, as in the more usual forms, to differences of density in the contents. Another effective water-tube boiler (Niclausse's) is composed of a group of Field tubes arranged so that the inner tube, through which the water flows on its way to the heating surface, opens out of one main drum or tube, while the outer member of each Field tube discharges steam and water into a second drum distinct from the first, but connected with it by a pipe through which the unevaporated water drains back.

The construction of water-tube boilers has received much attention at the hands of Mr J. I. Thornycroft, especially in relation to marine engines. In his form of boiler[1] (fig. 172, the entire heating surface is made up of tubes an inch or an inch and a half in diameter and therefore not requiring to be more than one-tenth of an inch thick for the working pressure of

[1] *Min. Proc. Inst. C. E.* Vol. xcix. p. 41. This paper also describes other forms and contains an important discussion of the whole subject.

200 or 250 lbs. The tubes form an arch over the fire and after bending out again terminate in the top of a separating drum

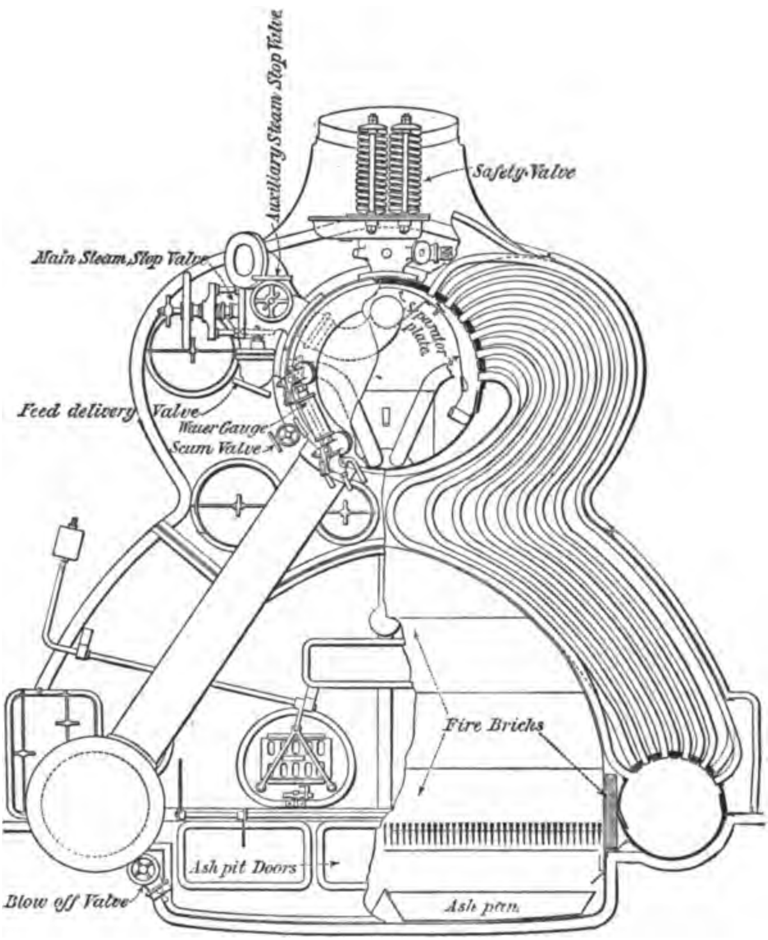


Fig. 172. Thornycroft Boiler.

from which the water drains by a pair of external pipes to the two drums which are seen at the base of the arch on either side. A boiler of this class fitted in a torpedo boat, with 1837 square feet of heating surface and 30 square feet of grate surface was tested by Professor Kennedy under various degrees of forced draught ranging up to a stokehole pressure of two inches of water[1].

[1] *Min. Proc. Inst. C. E.* Vol. xcix. p. 57.

Under the highest pressure of air it made enough steam to give 775 indicated horse-power in the engine; the heat used in making the steam was 67 per cent. of the whole energy of the fuel, and nearly 70 lbs. of coal were burnt per hour per square foot of grate. Analysis of the furnace gases showed that the supply of air per lb. of coal was 17·2 lbs., and that about 9 per cent. of the energy of the fuel was lost through imperfect combustion. In another trial when the air pressure in the stokehole was equivalent to only half-an-inch of water the engine gave 450 indicated horse-power, and with practically the same supply of air per lb. of coal 78 per cent. of the energy was used in making steam and only 5 per cent. was lost through imperfect combustion. These figures, and others which will be found in Professor Kennedy's report, show that a boiler of this kind can make steam with great freedom and with but little reduction in efficiency even when the draught is strongly forced, while its efficiency at more ordinary rates of output is remarkably high. Mr Yarrow's boiler is a somewhat similar form, the chief difference being that its tubes are straight and enter the separating drum below instead of above the water line. Another point of difference is that the Yarrow boiler has no external pipe to complete the circuit in which the water travels. The circuit is none the less complete, for some of the tubes between the separating drum and each of the two water-drums play the part of return tubes, through which the water streams down, while in others, more actively exposed to the fire, the current of mixed steam and water passes up. The tubes are in two groups, converging above on the separating drum, and forming a bridge over the grate[1].

221. Locomotive Boilers. The locomotive boiler consists of a nearly rectangular fire-box, enclosed above and on the sides by water, attached to a cylindrical part called the barrel, which extends horizontally from the fire-box to the front part of the locomotive and is filled with numerous horizontal tubes. Figs. 173 and 174 show in longitudinal and transverse section a boiler

[1] For a discussion of water-tube boilers see the *Transactions* of the Institution of Naval Architects, 1894. Reference should particularly be made to the experiments of Professor Watkinson on model boilers with glass tubes, exhibiting the characteristic action in boilers of the Babcock, Belleville, Thornycroft, Yarrow, Niclausse, and other types (*Trans. Inst. Nav. Arch.* 1896.)

of the London and North-Western Railway, which may be taken as typical of English practice.

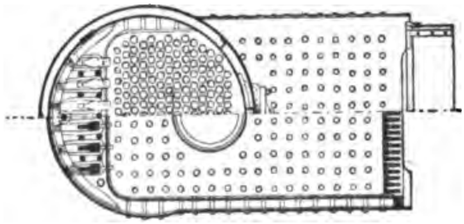


Fig. 174. Transverse section, half through fire-box and half through barrel.

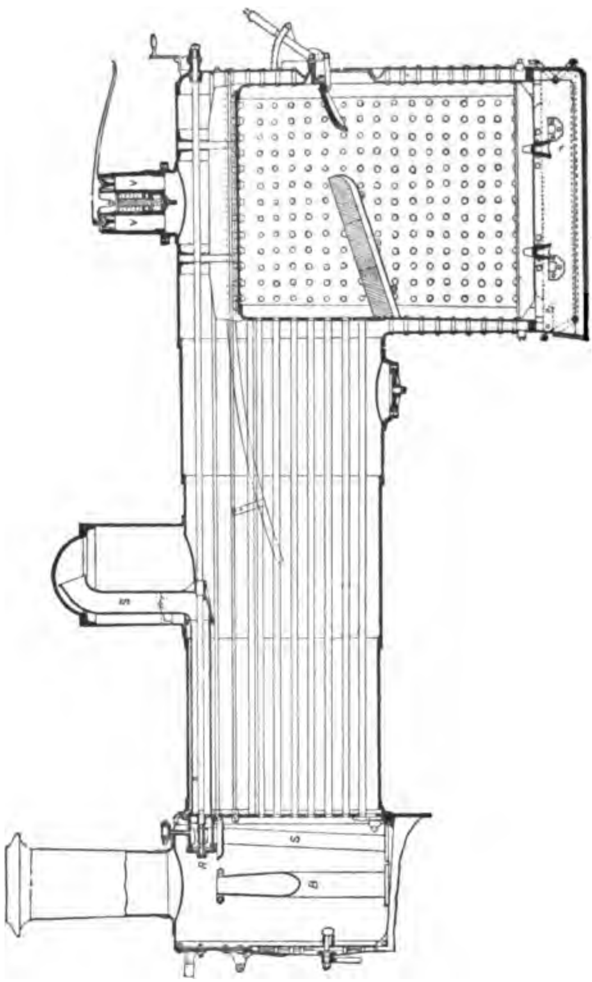


Fig. 173. Locomotive Boiler, longitudinal section.

The barrel is 10 feet long and a little more than 4 feet in diameter, and is made up of three rings of steel plates arranged telescopically. It contains 198 brass tubes, each $1\frac{7}{8}$ inches in external diameter. The front tube-plate in which the tubes terminate is of steel and is stayed to the back tube-plate by the tubes themselves, and the upper part of the front tube-plate above the tubes is also tied by longitudinal rods to the back end-plate. The fire-box is of copper and is nearly rectangular, with a horizontal grate. Round its sides, front, and back (except where the fire-door interrupts) is a water space about 3 inches wide, which narrows slightly towards the bottom. The flat sides of the fire-box are tied to the flat sides of the shell by copper stay-bolts, 4 inches apart, which are secured by screwing them into both plates and riveting over the ends. The crown of the fire-box is stiffened by a number of girders on the top, to which the plates are secured by short bolts. The girders are themselves hung from the top of the shell above them by slings which are secured to angle-irons riveted on the inside of the shell plates. A sloping bridge of fire-brick partially separates the upper part of the fire-box from the lower and prevents the flame from striking the tubes too directly. Under the grate is an ashpan, to which the supply of air is controlled by a damper in front. The fire-door opens inwards, and can be set more or less open, to regulate the amount of air admitted above the fire. On top of the barrel is a steam-dome, from which the steam supply is taken through a pipe *S* traversing the forward part of the steam space and passing down to the valve-chest through the smoke-box. The stop-valve or "regulator" *R* is situated in the smoke-box, and is worked by a rod through the boiler from the cab at the back. Above the fire-box end of the shell are a pair of Ramsbottom safety-valves, *V*, *V*—two valves pressed down by a single spring attached to the middle of a cross bar, which is prolonged to form a hand lever by which the valves may be eased up to see that they are free upon their seats. In front of the forward tube-plate is the smoke-box, containing a blast-pipe *B* by which the exhaust steam is used to produce a partial vacuum and so force a draught through the furnace.

Instead of stiffening the fire-box crown by the use of girder stays, the plan is sometimes followed of staying it directly to the shell above. The outer shell above the fire-box is generally

cylindrical; but to facilitate this method of staying it is sometimes made flat. This construction is not unusual in American locomotive boilers, another feature of which is that the grate is made larger than in English practice, for the purpose of burning anthracite coal. An extreme instance is furnished by the Wooton engines of the Philadelphia and Reading Railroad, which burn small coal of poor quality in a fire-box 9½ feet long by 8 feet wide, extending over the trailing wheels of the engine. In some cases the fire-box is divided by a sloping partition of plates with water between, which crosses the fire-box diagonally from front to back and has in its centre an opening resembling a fire-door mouthpiece to allow the products of combustion to pass. In others the fire-bridge is supported by water-tubes, and water-tubes are also used as grate-bars. This is done rather to promote circulation of the water than to give heating surface. The practice of American and English locomotive engineers differs somewhat as regards the materials of construction. American shells are of mild steel, English shells generally of mild steel but often of wrought-iron. In English practice the fire-boxes are of copper and the tubes of brass; in America the fire-boxes are of mild steel and the tubes of wrought-iron.

The locomotive type of boiler is used for portable and semi-portable engines, and to a considerable extent for stationary engines of small and medium power. It also finds a place in marine practice in cases where lightness is of special advantage.

222. **Marine Boilers.** So long as marine engines used steam of a pressure less than about 35 lbs. per square inch the marine boiler was generally a box with flat sides, elaborately stayed, with a row of internal furnaces near the bottom opening into a spacious combustion-chamber enclosed within the boiler at the back, and a set of return tubes leading from the upper part of the chamber to the front of the boiler, where the products of combustion entered the uptake and passed off to the funnel. The use of higher pressures has made this form entirely obsolete. The ordinary marine boiler is now a short circular horizontal cylinder of steel, closed by flat plates at the ends, with internal furnaces in cylindrical flues, internal combustion-chambers, and return tubes above the flues. This type is often described as the Scotch boiler. In one variety, called the double-ended boiler,

there are furnaces at both ends of the shell, each pair leading to a combustion-chamber in the centre that is common to both, or to separate central chambers with a water space between them.

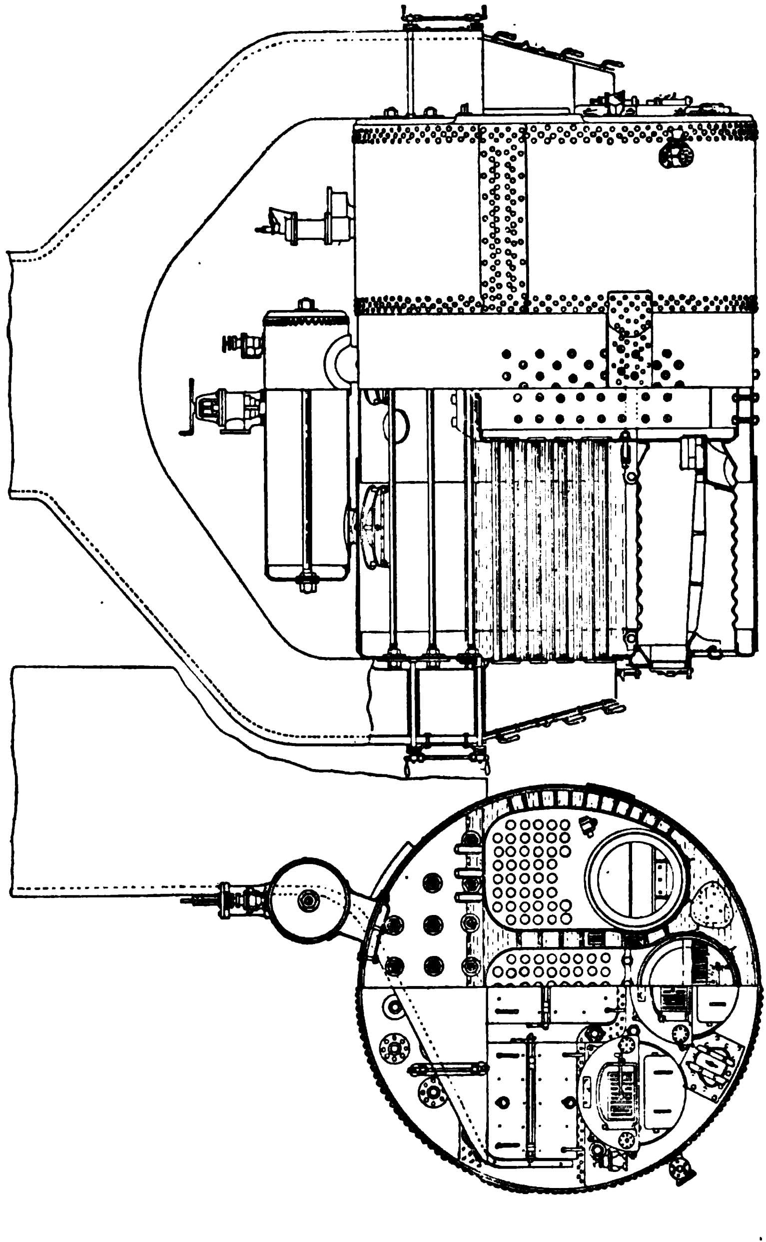


Fig. 176.

Fig. 175. Marine Boiler.

Figs. 175 and 176 show a double-ended marine boiler built by Messrs Gourlay Brothers for supplying steam at a pressure of 165 lbs. to a triple-expansion engine. At each end there are three furnaces in flues made of welded corrugated steel plates. The use of corrugated plates for flues, introduced by Mr Fox, makes thin flues able to resist collapse, and allows the flues to accommodate themselves easily to changes of temperature. One combustion-chamber is common to each pair of furnaces. It is strengthened on the top by girder stays and on the sides by stay-bolts to the neighbouring chamber and to the shell. The tubes are of iron, and a certain number of them are fitted with nuts so that they serve as stays between the tube-plate of the combustion-chamber and the front of the boiler. The upper part of the front plate is tied to the opposite end of the boiler by long stays. The uptakes from both ends converge to the funnel base above the centre of the boiler's length. The boiler shown is one of a pair, which lie side by side in the vessel, the uptake at each end being common to both. Each boiler in this example has a steam-dome, which is a part now often omitted, and from it the steam-pipe leads to the engine; it consists of a small cylindrical vessel, with flat ends tied together by a central stay. Short pipes connect the dome near each end with the steam space of the main shell. The shell is 12¼ feet in diameter, and 16½ feet long. The plates are of mild steel 1⅛ inches thick round the shell and 1 inch in the ends, the corrugated flues are ½ inch thick There are 127 tubes at each end, 46 of which are stay-tubes. The tubes are of iron, 3½ inches in external diameter. Above these are 18 longitudinal steel stay-rods extending from one end-plate to the others in the steam space. The crowns of the combustion-chambers are stiffened by girder stays, and their sides and bottom by short stay-bolts which tie them to one another and to the shell.

The single-ended marine boiler is practically half a double-ended boiler. The furnace doors are at one end only, and the boiler terminates in a flat end-plate which leaves only a few inches of water space between it and the back of the combustion-chambers, the end plate and the back plate of each chamber being tied together across this space by short stay-bolts.

Reference has already been made to the use on board ship of boilers of the locomotive and water tube types as substitutes for

these normal marine forms, the motive being to save weight, and also to use a higher pressure than can readily be borne by a large shell. Water-tube boilers, in particular, are now extensively used in war-ships of the largest size, as well as in torpedo destroyers and other small craft. Pressures of 250 lbs. per square inch and even more have consequently become common.

223. Feeding Boilers. The Injector. Boilers are usually fed either by a feed-pump driven by the engine, or by a distinct auxiliary engine called a "donkey," or by an injector. The *injector,* invented by the late M. Giffard, and now very generally used on locomotive and other boilers, is illustrated in fig. 177. Steam enters from the boiler at A and blows through an annular orifice B, the size of which is regulated by the handle C. The feed-water flows in at D, and meeting the steam at B causes it to

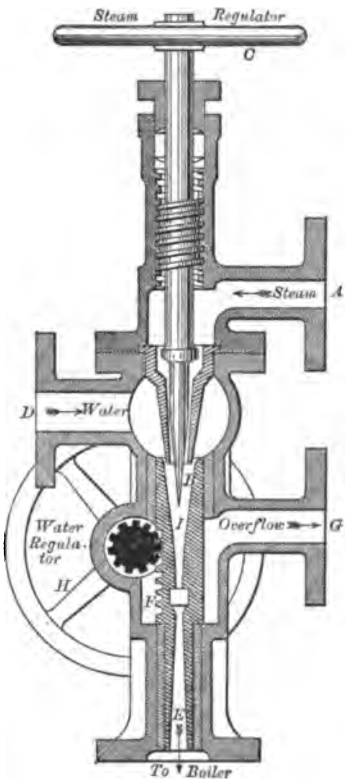


Fig. 177. Giffard's Injector.

condense. This produces a vacuum at B, and consequently the water rushes in with great velocity, and streams down through the combining nozzle I, its velocity being augmented by the impact of steam on the back of the column. In the lower part of the nozzle E the stream expands; it therefore loses velocity, and, by a well-known hydrodynamic principle, gains pressure, until at the bottom its pressure is so great that it enters the boiler through a check-valve which opens only in the direction of the stream. The orifice F which is in communication with the narrow end of the combining nozzle I and leads to the overflow pipe G allows the injector to start into action, by providing a channel through which steam and water may escape until the stream acquires enough energy to force its way into the boiler. The opening for admitting water between D and B is regulated by the wheel H. The form of injector shown in the sketch is substantially the original form introduced by Giffard: variations in it have been made by many makers. In some of these the injector is classified as *non-lifting*, that is to say, it requires the supply of water to come from a source at a level not lower than that of the injector. The non-lifting injector requires fewer adjustable parts: the steam-regulating cone B is omitted. The *exhaust-steam injector* works by steam from the exhaust of non-condensing engines, instead of using live steam from the boiler. The steam orifice is then larger in proportion to the other parts, the volume of the steam supply being greater. In *self-starting injectors* an arrangement is provided by which overflow will take place freely until the injector starts into action and then the openings are automatically adjusted to suit delivery into the boiler. One plan of doing this is to make the combining nozzle under the steam orifice in a piece which is free to slide in the outer casing. Until the injector starts it lies at some distance from the steam orifice, and allows free overflow; but when the vacuum forms it rises, in consequence of pressure at the base. In self-adjusting injectors this rise of the combining nozzle is made use of to contract the water-way round the steam orifice. In another form of self-starting injector one side of the combining nozzle is in the form of a hinged flap, which opens backwards to allow overflow to take place, but closes up when a vacuum is formed and the injector starts into action[1]. Weir's *hydrokineter* for large

[1] See papers in *Proc. Inst. Mech. Eng.* 1860, 1866, 1884. For the theory of the injector, see Peabody's *Thermodynamics of the Steam-Engine*, Chapter x.

marine boilers is another apparatus in which the principle of the injector is made use of, with the object of promoting circulation of the water during the time steam is being raised. It consists of a series of nozzles, with water-inlets between them, through which water is drawn by means of a central jet of steam supplied from a donkey boiler. The *ejector-condenser* of Mr Morton is another apparatus in which the principle of the injector is applied.

224. Feed-water heaters. In the boilers of factory and other stationary engines the use of a feed-water heater to extract from the furnace gases more heat than the heating surface of the boiler will itself take up, is common even when (as is generally the case) the chimney is relied on to maintain the draught. Green's "economizer" is a well-known form, in which the water passes through tubes the outer surface of which is exposed to the hot gases and kept clear of deposited soot by the continuous action of a mechanical scraper. A precisely similar construction has been made to serve as a superheater, and the two have been used together, the hot gases coming first into contact with the tubes of the superheater and then, at a slightly lower temperature, with those of the feed-water heater. In locomotives and other non-condensing engines a portion of the exhaust steam is frequently made use of to heat the feed-water. When an exhaust-steam injector is employed it serves the purpose of a feed-water heater as well as that of a feed-pump. Besides increasing the efficiency of the boiler by utilizing what would otherwise be waste heat, a feed-water heater has the advantage that by raising the temperature of the water it removes air, and also, in the case of hard water, causes lime and other substances held in solution to be deposited in the heater instead of being carried into the boiler, where they would form scale. In Weir's feed-heater for marine engines the temperature of the feed-water is raised to about 200° Fahr. by injecting steam from the intermediate receiver. When a donkey pump is used for feeding boilers the feed is often heated by allowing the steam used by the donkey to be condensed in it.

225. Use of Zinc to prevent corrosion in boilers. To prevent corrosion in boilers it is usual to introduce blocks of zinc in metallic connexion with the shell. These are set in the water space, preferably at places where corrosion has been found specially liable to occur. Their function is to set up a galvanic action,

in which zinc plays the part of the negative element, and is dissolved while the metal of the shell is kept electro-positive. Otherwise there would be a tendency for differences of electric quality between different parts of the shell to set up galvanic actions between the parts themselves, by which some parts, being negative to others, would be attacked. The zinc raises the potential of the whole shell enough to make all parts positive relatively to the water.

226. Methods of forcing draught. The simplest but by no means the most economical way of forcing the draught is to let a jet of steam from the boiler discharge itself up the chimney. It tends to carry the furnace gases with it and so to reduce the air pressure in the space where the jet escapes. Allusion has already been made to the system which is universal in locomotive boilers of utilizing the exhaust steam from the engine as a means of forcing the draught. Two methods of mechanically forcing the draught have come into extensive use in marine practice. One plan is to box in the stokehole and keep the air in it at a pressure of two or three inches of water, or occasionally more, by the use of blowing fans. In Mr Howden's system of forced draught the stokehole is open, and air is supplied by a blowing fan to a reservoir formed by enclosing the ashpit and also to another reservoir from which air gets access to the grate above and through the fire-door. On its way to the reservoir the air is heated by passing across a part of the uptake in which the hot gases from the furnace are led through tubes. This method of restoring to the furnace what would otherwise be waste heat forms an interesting alternative to the method of restoring heat to the boiler by passing the hot gases through a feed-water heater; it is in fact an application to boiler furnaces of the regenerative principle alluded to in Chap. II.[1]

By either of these means the power of the boiler is increased in the ratio of 3 to 2, or even more, as compared with its power under chimney draught. The efficiency of the boiler is, in general, slightly but not very materially reduced. Arrangements are often made which allow the chimney draught to serve in ordinary steaming and the fan to be resorted to when an exceptional demand for power has to be met: in other cases the pressure at

[1] A description and discussion of these alternative methods of forcing draught will be found in papers read before the Institution of Naval Architects, April 1886.

which air is supplied and consequently the rate of combustion of fuel on the grate is regulated by varying the speed of the fan. The facility which forced draught offers for adapting a boiler to a wide range of power has been illustrated in the experiments quoted in § 220. An ordinary marine boiler burns 15 to 20 lbs. of coal per hour per square foot of grate with natural draught, and this is easily raised to 30 lbs. or more under forced draught. A locomotive using the steam blast will burn 70 or 80 lbs. per hour per square foot of grate, and in boilers of the locomotive type which have been employed in torpedo boats a consumption at the rate of 140 lbs. has been reached. In such cases however the efficiency is low, for the combustion is not very perfect and the temperature of the escaping gases is high.

227. Mechanical Stoking. Many appliances have been devised for the mechanical supply of coal to boiler furnaces, but these have hitherto taken the place of hand-firing to only a very limited extent. In Juckes's furnace the fire-bars are in short lengths, jointed by pins to form a continuous chain or web, which rests on rollers and is caused to travel slowly in the direction of the furnace's length by pin-wheels round which the web is carried at the front and back. Coal is allowed to drop continually on the travelling grate from a hopper in front of the furnace. A more usual form of mechanical stoker is a reciprocating shovel or ram which is supplied with coal from a coal-hopper, and throws or pushes a small quantity of coal into the fire at each stroke. Along with this devices are employed for making the grate self-cleansing, by giving alternate fire-bars a rocking or sliding motion through a limited range. In Mr Crampton's dust-fuel furnace the coal was ground to powder and fed by rollers into a pipe from which it was blown into the furnace by an air-blast. This gave so intimate a mixture of fuel and air that the excess of air required for dilution was only one-fifth of the amount required for combustion[1]. A similar advantage attends the use of gaseous fuel, and of liquid fuel that is blown into the furnace in the form of spray.

228. Liquid Fuel. The use of liquid fuel for boilers has acquired considerable importance, mainly in connexion with the discovery of petroleum, in large quantity, at Baku on the Caspian

[1] *Proc. Inst. Mech. Eng.* 1869.

Sea. The "astatki" or heavy petroleum refuse which is left after distilling paraffin from the crude oil forms an exceedingly cheap fuel, with a calorific value from one-third to one-half greater than that of an equal weight of coal. It has superseded coal in the steamers of the Caspian, and has been very largely employed for locomotives in the south-eastern part of Russia. The oil is injected in the form of spray near the foot of the fire-box by a steam jet, which is arranged in such a way that air is drawn into the furnace along with the petroleum. In the arrangement for burning petroleum introduced in Russian locomotives by Mr T. Urquhart the flame impinges on a structure of fire-brick, built in the fire-box with numerous openings to allow the gases to diffuse themselves throughout the combustion-chamber. This guards against a too intense play of flame on the metallic surfaces, and at the same time the bricks serve as a reservoir of heat to rekindle the flame should the combustion be intermittent. In getting up steam an auxiliary boiler is used to supply the jet which serves to convert the oil into spray and to inject it along with air into the furnace[1]. Obvious advantages of liquid fuel are the ease with which the rate of combustion can be regulated to suit sudden changes in the demand for steam, also the nicety with which the supply of air taken in by the oil injector can be adjusted, and the continuity with which the fuel is supplied, its entrance requiring no opening of the fire-door with a consequent inrush of cold air.

Owing to the cost and danger of transporting oil in bulk its use as furnace fuel, although perfectly successful in the neighbourhood of the oil wells, has hitherto been almost wholly restricted to certain districts. In England however Mr Holden has used with success not only Russian petroleum refuse but also the residuum from shale-oil distilleries with tar and other heavy liquid refuse as a substitute for coal in a number of the locomotives of the Great Eastern Railway. In some cases the oil is used alone; in others it is used in conjunction with coal. It is sprayed into the fire-box by steam injectors, mixing in the injector with air which has been previously heated to about 300° Fah. by passing through a heater in the smoke-box. When oil is used alone the consumption per mile is stated to be 16·5 lbs. as against 35·4 lbs. of coal. Mr Aspinall also uses oil fuel in locomotives of the Liverpool Dock

[1] Urquhart, *Min. Proc. Inst. C. E.* 1884.

lines, mainly to reduce the risk of fire from sparks and to prevent smoke[1].

229. Superheating. The most usual mode of superheating steam is to cause it to traverse a stack of pipes which is placed in the boiler flue, or, in exceptional cases, is separately fired. When set in the boiler flue the superheater should be so placed that the furnace gases act on it before their heat has been extracted by passing over all the heating surface of the boiler. If placed in the uptake of the flues the superheater will be comparatively ineffective, especially when the boiler is steaming lightly. In Schwörer's superheater, a form largely used in Alsace (where superheating has been commonly practised ever since its advantages were established by the experiments of Hirn), the steam passes through a single short coil of comparatively large pipe which has gills cast on it to enlarge its heating surface. Gehre's superheater consists of a pair of cylindrical drums between the end plates of which are fitted a large number of straight tubes. Steam circulates through the drum, while the hot gases pass through the tubes as well as along the outside of the drums. Another form of superheater is made by having a group of U-shaped tubes in the flue with their ends fitted into a box or pipe with a partition which compels the steam to go down one leg of each U and up the other[2].

230. Steam Boiler Trials. For examples of complete boiler trials reference should be made to the series of tests conducted by Messrs Bryan Donkin and Kennedy and published in detail in *Engineering* from 1890 to 1894. These trials, twenty-one in number, deal with boilers of nine different types and in most cases include analyses of the furnace gases. From these analyses, along with measurements of the coal burnt and of the water evaporated, heat accounts are made out showing the items which compose the total expenditure of heat, and the quantity of air passing through the furnace is determined. From a summary given by Mr Donkin

[1] Aspinall on Petroleum as Steam-Engine Fuel. Institution of Civil Engineers, Conference, 1897.

[2] For particulars of various superheaters see Mr Patchell's "Notes on Steam Superheating," *Proc. Inst. Mech. Eng.* April 1896. The discussion on this paper gives much interesting information on the general question of the use of superheated steam.

TABLE XII. *Results of Boiler Trials.*

Type of Boiler.	Cornish.			Lancashire.		Locomotive (Stationary type).			Vertical tubular.	Fire-Engine (vertical tubular).	Water-tube.
Ratio of heating surface to area of grate.	19	33	54	39	45*	31	33	37	74	45	31*
lbs. of coal used per hour per square foot of grate.	15·7	6·9	13·7	10·2	16·8	6·2	12·4	25·5	17·9	34·3	14·3
lbs. of water evaporated (from and at 212°) per hour per square foot of heating surface.	7·4	2·3	2·5	2·6	4·2	1·7	4·0	7·2	2·3	13·1	4·5
lbs. of water evaporated (from and at 212°) per lb. of coal.	9·16	11·40	9·91	9·92	12·46	8·45	10·78	10·64	9·57	7·95	11·35
lbs. of air per lb. of coal.	28	21	28	24		41	19		16		29
Percentage of energy in fuel utilized in the boiler in heating and evaporating the water.	62	78	65	70	71	57	70	73	64	51	62
Ditto in the economiser.					10						12

of the results of the experiments[1] a number of more or less representative figures have been selected (Table XII). In the two trials marked with an asterisk an economiser was used to heat the feed water, but the stated area of heating surface is that of the boiler alone and does not include the supplementary surface furnished by the economiser. The stated values of the evaporated water are not the actual quantities heated and evaporated under pressure, but the equivalent quantities from and at 212° Fah.

Results are also stated for a locomotive tested while running, but the heat account shows that priming has probably interfered with their accuracy. In the trials of stationary locomotive boilers it is interesting to notice how the efficiency is reduced by a too free admission of air, in the first of the three experiments cited here.

[1] *Engineering*, Sept. 20, 1895.

CHAPTER XII.

FORMS OF THE STEAM-ENGINE.

231. Terms used in classification. In classifying engines with regard to their general arrangement of parts and mode of working, account has to be taken of a considerable number of independent characteristics. We have, first, a general division into *condensing* and *non-condensing engines*, with a subdivision of the condensing class into those which act by surface condensation and those which use injection. Next there is the division into *compound* and *non-compound*, with a further classification of the former as double-, triple-, or quadruple-expansion engines. Again, engines may be classed as *single* or *double-acting*, according as the steam acts on one or alternately on both sides of the piston. Again, a few engines—such as steam-hammers and certain kinds of steam-pumps—are *non-rotative*, that is to say, the reciprocating motion of the piston does work simply on a reciprocating piece; but generally an engine does work on a continuously revolving shaft, and is termed *rotative*. In most cases the crank-pin of the revolving shaft is connected directly with the piston-rod by a connecting-rod, and the engine is then said to be *direct-acting*; in other cases, of which the ordinary beam-engine is the most important example, a lever is interposed between the piston and the connecting-rod. The same distinction applies to non-rotative pumping engines, in some of which the piston acts directly on the pump-rod, while in others it acts through a beam. The position of the cylinder is another element of classification, giving *horizontal*, *vertical*, and *inclined cylinder* engines. Many vertical engines are further distinguished as belonging to the *inverted cylinder* class; that is to say, the cylinder is above the connecting-rod and crank.

In *oscillating cylinder engines* the connecting-rod is dispensed with; the piston-rod works on the crank-pin, and the cylinder oscillates on trunnions to allow the piston-rod to follow the crank-pin round its circular path. In *trunk engines* the piston-rod is dispensed with; the connecting-rod extends as far as the piston, to which it is jointed, and a trunk or tubular extension of the piston, through the cylinder cover, gives room for the rod to oscillate. In *rotary* engines there is no piston in the ordinary sense; the steam does work on a revolving piece, and the necessity is thus avoided of afterwards converting reciprocating into rotary motion. Steam *turbines* may be said to form an extreme development of this rotary class. Still another mode of classification speaks of engines in reference to the conditions under which they are to work, as *stationary, locomotive*, or *marine*.

232. Beam-Engines. In the single-acting atmospheric engine of Newcomen the beam was a necessary feature; the use of water-packing for the piston required that the piston should move down in the working stroke, and a beam was needed to let the counterpoise pull the piston up. Watt's improvements made the beam no longer necessary; and in one of the forms he designed it was discarded—namely, in the form of pumping-engine known as the Bull engine, in which a vertical inverted cylinder stands over and acts directly on the pump-rod. But the beam type was generally retained by Watt, and for many years it remained a favourite with the builders of large engines. The beam formed a convenient driver for pump-rods and valve-rods; and the parallel motion which had been invented by Watt as a means of guiding the piston-rod could easily be applied to a beam-engine, and was, in the early days of engine-building, an easier thing to construct than the plane surfaces which are the natural guides of the piston-rod in a direct-acting engine. In modern practice the direct-acting type has almost wholly displaced the beam type. For mill-driving and the general purposes of a rotative engine the beam type is now rarely chosen. In pumping engines it is more common, but even there the tendency is to use direct-acting forms.

The only distinctive feature of beam-engines requiring special notice here is the "parallel motion," an ordinary form of which is shown diagrammatically in fig. 178. There *MN* is the path in which

the piston-rod head, or crosshead, as it is often called, is to be guided. ABC is the middle line of half the beam, C being the fixed centre about which the beam oscillates. A link BD connects a point in

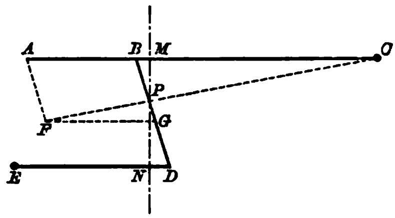


FIG. 178.—Watt's Parallel Motion.

the beam with a radius link ED, which oscillates about a fixed centre at E. A point P in BD, taken so that $BP:DP::EN:CM$, moves in a path which coincides very closely with the straight line MPN. Any other point F in the line CP or CP produced is made to copy this motion by means of the links AF and FG, parallel to BD and AC. In the ordinary application of the parallel motion a point such as F is the point of attachment of the piston-rod, and P is used to drive a pump-rod. Other points in the line CP produced are occasionally made use of, by adding other pairs of links parallel to AC and BD.

Watt's linkage gives no more than an approximation to straight-line motion, but in a well-designed example the amount of deviation need not exceed one four-thousandth of the length of the stroke. It was for long believed that the production of an exact straight-line motion by pure linkage was impossible, until the problem was solved by the invention of the Peaucellier cell. The Peaucellier linkage has not been applied to the steam-engine, except in isolated cases.

In by far the greater number of modern steam-engines the crosshead is guided by sliding on planed surfaces. In many beam-engines, even, this plan of guiding the head of the piston-rod has taken the place of the parallel motion.

233. Direct-acting Horizontal and Vertical Engines. This plan of guiding the piston-rod head is practically universal in engines of the direct-acting class: the piston, the connecting-rod and the crank constitute a "slider-crank-chain" with the frame or bed-plate of the engine to form the fourth element.

No type of steam-engine is so common as the horizontal direct-acting. In small forms the engine is generally self-contained, in other words, a single frame or bed-plate carries all the parts, including the main bearings in which the crank-shaft with its fly-wheel turns. The cylinder either rests on the bed-plate, or overhangs at the back, being in the latter case bolted to a vertical part of the frame which forms a cover for the front end of the cylinder. The frame is often given what is called a girder shape, which brings a portion of it more directly into the line of thrust between the cylinder and the crank centre, and allows the upper as well as the lower of the two surfaces which serve as guides for the crosshead to be formed on the frame itself. This construction is found in many small engines and is also usual in large engines of the Corliss type. The feed-pump plunger is usually driven from a separate excentric: in some cases it is directly attached to the crosshead, and in others to the valve-rod. When a condenser is used with a small horizontal engine it is usually placed behind the cylinder, and the air-pump, which is within the condenser, is a horizontal plunger or piston-pump worked by a 'tail-rod'—that is, a continuation of the piston-rod past the piston and through the back cover of the cylinder. In large horizontal engines the condenser generally stands in a well between the cylinder and the crank-shaft, and the pump, which has a vertical stroke, is worked by means of a bell-crank lever attached by a link to the crosshead of the engine.

When uniformity of driving effort or the absence of dead points is specially important, two independent cylinders often work on the same shaft by cranks at right angles to each other, an arrangement which allows the engine to be started readily from any position. Such engines are called *coupled.* The ordinary locomotive is an example of this form. Winding engines for mines and collieries, in which ease of starting, stopping, and reversing is essential, are very generally made by coupling a pair of horizontal cylinders, with cranks at right angles to each other, on opposite sides of the winding-drum, with the link-motion as the means of operating the valves.

Direct-acting engines of the larger class are generally compounded either (1) by having a high and a low pressure cylinder side by side, working on two cranks at exactly or nearly right angles to each other, or (2) by placing one cylinder behind the

other, with the axes of both in the same straight line. The latter is called the *tandem* arrangement. In it one piston-rod is generally common to both cylinders; occasionally, however, the piston-rods are distinct, and are connected to one another by a framing of parallel bars outside of the cylinders. Another construction, rarely followed, is to have parallel cylinders with both piston-rods acting on one crank by being joined to opposite ends of one long crosshead. In a few compound engines the large cylinder is horizontal, and the other lies above it in an inclined position, with its connecting-rod working on the same crank-pin.

In tandem engines, since the pistons move together, there is no need to provide a receiver between the cylinders. It is practicable to follow the "Woolf" plan (§ 146) of allowing the steam to expand directly from the small into the large cylinder; and in many instances this is done. In any case, however, the connecting-pipe and steam-chest form an intermediate receiver of considerable size, which will cause loss by "drop" (§ 148) unless steam be cut off in the large cylinder before the end of the stroke. Hence it is more usual to work with a moderately early cut-off in the low-pressure cylinder than to admit steam to it throughout the whole stroke. Unless it is desired to make the cut-off occur before half-stroke, an ordinary slide-valve will serve to distribute steam to the large cylinder. For an earlier cut-off than this a separate expansion-valve is required on the low-pressure cylinder, to supplement the slide-valve; and in any case, by providing a separate expansion-valve, the point of cut-off is made subject to easy control, and may be adjusted so as to avoid drop or to divide the work as may be desired between the two cylinders. For this reason it is not unusual to find an expansion-valve, as well as a common slide-valve, on the low-pressure cylinder even in tandem engines. In many cases, however, the common slide-valve only is used. In the high-pressure cylinder of compound engines, the cut-off is usually effected either by an expansion slide-valve or by some form of Corliss or other trip-gear.

For mill engines the compound tandem and compound coupled types are the most usual, and the high-pressure cylinder is very generally fitted with Corliss gear. In the compound coupled arrangement the cylinders are on separate bed-plates, and the fly-wheel is between the cranks. The use of triple expansion

engines in mill-driving is extending but is still comparatively uncommon.

The general arrangement of vertical engines differs little from that of horizontal engines. The cylinder is usually supported above the shaft by a cast-iron frame resembling an inverted Λ, whose sides are kept parallel for a part of their length to serve as guides for the crosshead. Sometimes one side of the frame only is used, and the engine is stiffened by one or more wrought-iron columns between the cylinder and the base on the other side. *Wall-engines* are a vertical form with a flat frame or bed-plate, which is fixed by being bolted against a wall; in these the shaft is generally at the top. Vertical engines are compounded, like horizontal engines, either by coupling parallel cylinders to cranks at right angles (or at 120° if triple expansion is to be used, as in the ordinary marine form) or, tandem fashion, by placing the high-pressure cylinder above the other. In vertical condensing engines the condenser is situated near the base under the back limbs of the frame, and the air-pump, which has a vertical stroke, is generally worked by a horizontal lever connected by a short link to the crosshead. In some cases the pump is horizontal, and is worked by a crank on the main shaft[1].

In land engines using condensation the jet form of condenser is the most usual, surface condensation being resorted to only in special cases, as when the available supply of water is unsuited for boiler feed. When there is no large supply of condensing water a very fair vacuum can be obtained by using an *evaporative condenser*, consisting of a stack of pipes into which the exhaust steam is admitted and over which a comparatively small amount of cold water is allowed to drip. Such a condenser is placed in the open air, generally on the roof, and the amount of water used by it need not exceed the amount that is condensed. It can therefore be applied, to what would otherwise be non-condensing engines, giving the thermodynamic advantage of condensation without any additional expenditure of water, the feed water being saved by condensation while the quantity of cooling water is no more, and may even be less, than the quantity would be required for feed.

[1] For particulars of the usual forms taken by engines, see Haeder's *Handbook of the Steam-Engine*, Tran. by H. H. P. Powles (1893). The construction of details is discussed in Unwin's *Elements of Machine Design*, Vol. ii.

234. Single-acting High-speed Engines. The ordinary double-acting engine, whether of the horizontal or vertical variety, may be adapted to a high speed by lightening its reciprocating parts, enlarging its bearing surfaces, and taking pains to secure symmetry and balance in the design. Mr Thornycroft, for example, has reached a speed of 1000 revolutions per minute in the engines built by him to drive fans for forcing the draught of torpedo boats. And even the comparatively large engines of a thousand horse-power or more which drive the propellers in these vessels are made to run at 400 or 500 revolutions per minute, mainly through the use of exceptionally large bearing surfaces. Several successful instances of the double-acting high-speed type might be named, but the high-speed engines most usually met with are of the single-acting type. Steam is admitted to the back of the piston only, and the connecting-rod is in compression throughout the whole revolution. Besides simplifying the valves, this has the important advantage that alternation of strain at the joints may be entirely avoided, with the knocking and wear of the brasses which it is apt to cause. To secure, however, that the connecting-rod shall always push, there must be much cushioning during the back or exhaust stroke, for reasons which have been explained in Chapter X. From a point near the middle of the back stroke to the end the piston is being retarded; cushioning must begin at that point or earlier, and the work spent upon the cushion must at every stage be at least as great as the loss of energy on the part of the piston and rods. In some single-acting engines this cushioning is done by compressing a portion of the exhaust steam; in others the rod is kept in compression by help of a supplementary piston, on which steam from the boiler presses; in the Willans engine the cushioning is done by compressing air. The demand for an engine which should run fast enough to drive a dynamo directly, without a belt or other intermediate gearing to multiply the speed, has done much to bring engines of this class into use.

An early example is the three-cylinder engine introduced by Mr Brotherhood in 1873, a recent form of which is shown in figs. 179 and 180. Fig. 179 is a longitudinal and fig. 180 a transverse section. Three cylinders, set at 120° apart, project from a closed casing, the central portion of which communicates with the exhaust. The pistons have the trunk form—that is to say,

there is a joint in the piston itself which allows the piston-rod to oscillate, and so makes a separate connecting-rod unnecessary.

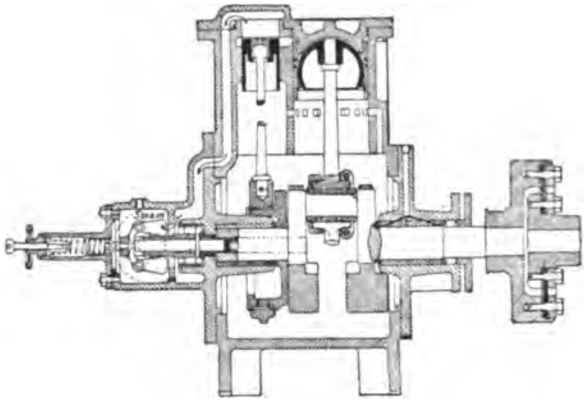


FIG. 179.—Brotherhood's Three-Cylinder Engine: longitudinal section.

The three rods work on a single crank-pin, and the balance weights are on a pair of crank cheeks on the other side of the shaft. Steam is admitted to the back of the pistons only. It passes first through a throttle-valve, which is controlled by a centrifugal spring-governor (fig. 179), and is then distributed to the cylinders by three piston-valves worked by an eccentric, the sheave of which is made hollow so as to overhang one of the main bearings. Release takes place by the piston itself uncovering exhaust ports in the circumference of the cylinder, and the rocking motion of the piston-rod is taken advantage of to open a

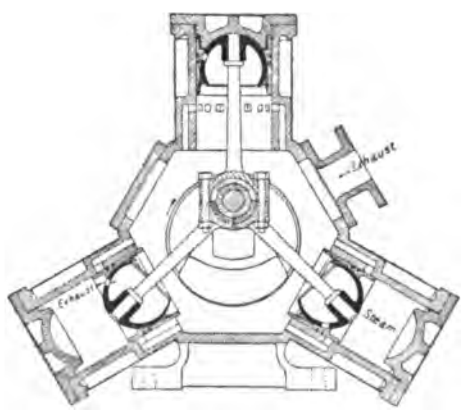


FIG. 180.—Brotherhood's Three-Cylinder Engine: transverse section.

supplementary exhaust port (fig. 180), which remains open during a sufficient portion of the back-stroke. The flexible coupling shown at the right-hand end of the shaft in fig. 179 transmits the twisting moment of the shaft through disks of leather, and so prevents straining of the shaft and bearings through any want of alignment between the shaft of the engine and that of the mechanism it drives. Besides its use as a steam-engine, Mr Brotherhood's pattern has been extensively applied in driving torpedoes by means of compressed air. As a steam-engine it is compounded by placing a high-pressure cylinder outside of and tandem with each of the three low-pressure cylinders.

In other single-acting engines the cylinders are placed side by side above the shaft, to act on two or on three cranks. The cranks and connecting-rods are completely enclosed, and are lubricated by dipping into a mixture of oil and water with which the lower part of the casing is filled. In the Westinghouse engine there are two vertical cylinders to which steam is admitted by piston-valves, and the crank-shaft is situated half a crank's length out of the line of stroke, to reduce the effect of the connecting-rod's obliquity during the working stroke.

To this type also belongs the Willans "central valve" engine, which has been repeatedly referred to in Chapter V. in connexion with the late Mr Willans' important experimental work. The exceptional efficiency of this engine in regard to consumption of steam, together with the facility it gives for driving a dynamo direct, and the small bulk which is a consequence of the high speed, has led to its being extensively used at electric lighting stations and elsewhere, in sizes ranging up to about 500 horse-power. In the compound and triple forms the successive cylinders are set tandem, in a vertical line, and the space below the upper piston serves as intermediate receiver. In some cases a single crank is used, but generally two or three sets of cylinders are grouped in parallel lines above a corresponding number of cranks. The advantage of three sets, acting on three cranks 120° apart, in respect of freedom from vibration has been pointed out in § 211. The piston-rod of each set of cylinders is hollow, and has a piston-valve in it, worked by an eccentric on the crank-pin, and arranged so that the relative movement of this valve with respect to the hollow rod determines the admission, transfer, and exhaust of the steam. The crosshead is itself a piston, working in a hollow

cylindrical guide, which becomes closed during the up-stroke so that air may be compressed in it to serve as a cushion and prevent the stress at the crank-pin from ever changing from push to pull, as was explained in § 209. The work stored in this air-cushion during the up-stroke is restored during the down-stroke, almost without loss. The eccentric rod which works the valve is also as a rule kept in compression by the pressure of the live steam

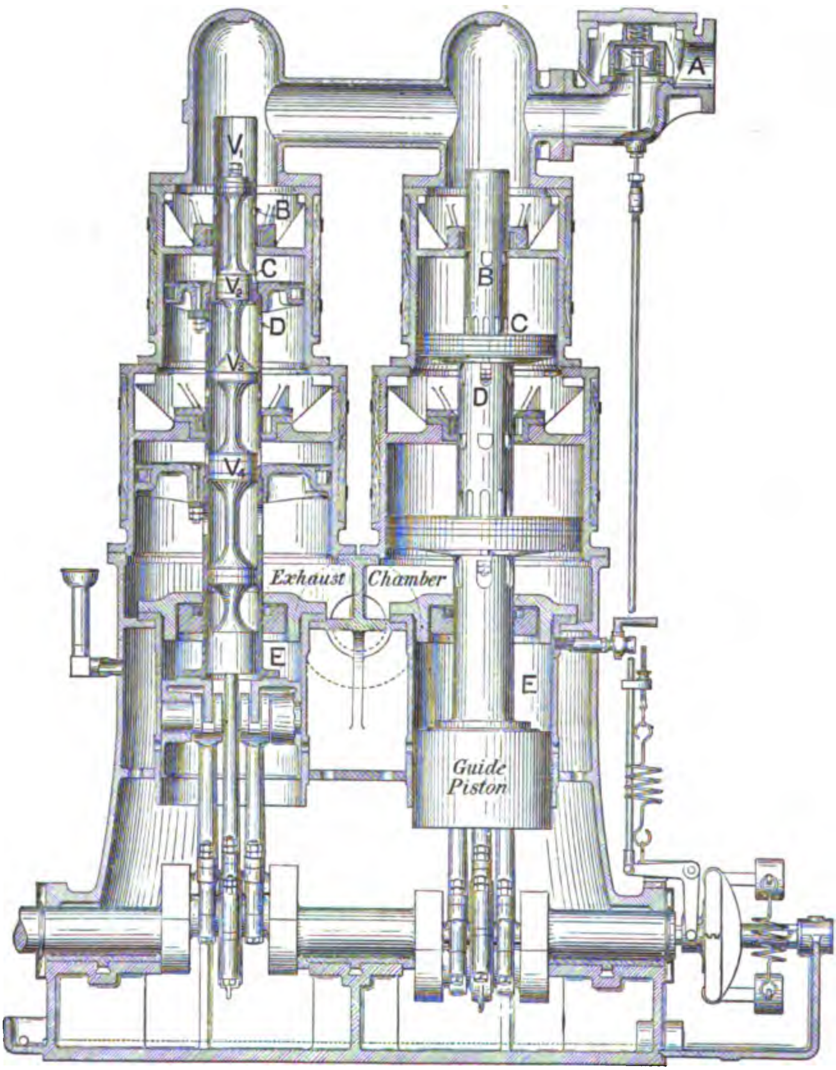


FIG. 181. Willans' Central Valve Engine.

on the topmost piston-valve. The rod is split up into two parallel parts and the valve eccentric is set between them, its sheave being forged on the crank-pin so that the relative motion of the valve to the piston-rod which encloses it may be the same as the motion which a valve in an ordinary engine performs relatively to its fixed seat when the valve is moved by an eccentric on its shaft. These features will be better understood by reference to fig. 181, which gives a section through the two lines of cylinders in a two-crank compound engine. The two lines are alike, but the drawing shows the piston-rod in section on the left-hand side, in order to let the piston-valves inside it be seen. Steam enters the steam-chest at the top through a double beat throttle-valve *A*, which is controlled by a centrifugal governor. It is admitted to the topmost cylinder through the ports *B* and *C* in the hollow piston-rod or "trunk." Cut-off occurs when the ports *B* are covered by disappearing into the gland of the top cylinder cover as the piston-rod descends. Subsequently the relative motion of the valve rod within makes the valve V_2 cover the port *C*, and (near the end of the down-stroke) puts the ports *C* and *D* in communication with each other between the valves V_2 and V_3, in order to allow the steam which has expanded in the first or upper cylinder to pass into the space between the upper piston and the cover of the second cylinder below. During the back-stroke the steam passes into this space, which serves as intermediate receiver, and thence is admitted, in the next down-stroke, into the second cylinder in just the same way as it was admitted into the first. For triple expansion a third cylinder is placed below the second, and the steam passes through it in the same way during a third revolution of the engine. It finally escapes to the exhaust chamber, the position of which is marked on the figure. The part of the stroke at which cut-off occurs is regulated by adjusting the height of the glands in the top of each cylinder cover. The air buffer is contained in the guide cylinders *EE*.

Several other forms of high-speed single-acting engines resemble the Willans engine in being completely encased and in effecting lubrication by making the cranks splash about oil and water inside the case. In Mather and Platt's form the connecting rods and piston-rods are kept in tension, steam being admitted only below the pistons. A supplementary balance piston, on the under side of which the steam presses continuously, serves to

maintain a state of tension in each rod when it would otherwise change to a state of compression in consequence of the inertia of the reciprocating parts (see § 209).

235. Pumping Engines. In engines for pumping water and other liquids, or for blowing air, it is not essential to drive a revolving shaft, and in many forms the reciprocating motion of the steam-piston is applied directly or through a beam to produce the reciprocating motion of the pump-piston or plunger without the intervention of any revolving part. On the other hand, pumping and blowing engines are frequently made rotative for the sake of adding a fly-wheel. When the level of the suction water is sufficiently high, horizontal engines, with the pump behind the cylinder and in line with it, are generally preferred; in other cases a beam-engine or vertical direct-acting engine is more common. Horizontal engines are, however, employed to pump water from any depth by using triangular rocking frames, which serve as bell-crank levers between the horizontal piston and vertical pump-rods. For deep-well or mine pumping the Cornish type still finds employment with its single cylinder, single action, and cataract control. The non-rotative engine frequently takes a double-acting and compound form, as for instance in fig. 182, which shows a compound inverted vertical

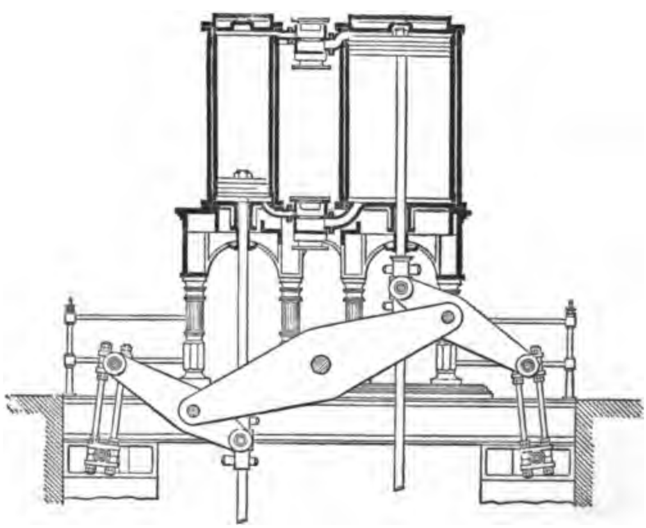


FIG. 182. Vertical Non-Rotative Pumping Engine.

pumping-engine of this class, by Messrs Hathorn, Davey and Co. Steam is distributed through lift valves, and the engine is governed by the differential gear illustrated in fig. 144, in conjunction with a cataract, which makes the pistons pause at the end of each stroke. The pistons are in line with two pump-rods, and are coupled by an inverted beam which gives guidance to the cross-heads by means of a link-work which produces an approximate straight-line motion.

Engines of this kind, like the old Cornish pump, are able to work expansively in consequence of the great inertia of the reciprocating pieces, the chief of which are the long and massive pump-rods. Notwithstanding the comparatively low frequency of the stroke, enough energy is stored in the movement of the rods to counterbalance the inequality with which the expanding steam works in different portions of the stroke, and the rate of acceleration of the system adjusts itself to give, at the plunger end, the nearly uniform effort which is required in the pump. In other words, the motion (instead of being almost simply harmonic as it is in a rotative engine) is such that the form of the inertia curve, when drawn as in fig. 149, is nearly the same as that of the steam curve, with the result that the distance between the two, which represents the effective effort on the pump-plunger, is nearly constant. The massive pump-rods may be said to form a reciprocating fly-wheel.

It is however only to deep-well pumping that this applies, and a very numerous class of direct-acting non-rotative steam-pumps have too little mass in their reciprocating parts to allow such an adjustment to take place at any ordinary speed. A familiar example is the small donkey pump used for feeding boilers, which has its steam-piston and pump-plunger on the same piston-rod. In such engines an auxiliary rotative element is often introduced, partly to secure uniformity of motion and partly for convenience in working the valves; a connecting rod, for instance, is sometimes taken from a point in the piston-rod to a crank shaft which carries a fly-wheel, or a slotted crosshead is fixed to the rod and gives motion of rotation to a crank-pin which gears in the slot, the line of the slot being perpendicular to that of the stroke. But many pumps of this class are purely non-rotative, and in such cases the steam is generally admitted throughout the whole of the stroke, since the inertia of the parts is not

sufficient to give the means of reconciling uniformity of pump-effort with expansive working. In some of these the valve is worked by tappets from the piston-rod. In the Blake steam-pump a tappet worked by the piston as it reaches each end of its stroke throws over an auxiliary steam-valve, which admits steam to one or other side of an auxiliary piston carrying the main slide-valve. In Cameron and Floyd's form one of a pair of tappet-valves at the ends of the cylinder is opened by the piston as it reaches the end of the stroke, and puts one or other side of an auxiliary pistón, which carries the slide-valve, into communication with the exhaust, so that it is thrown over. Often the working of the valve is secured by using a "duplex" pair of cylinders, and arranging them so that the motion of one controls the valve of the other. In the Worthington steam-pump, for example, two steam-cylinders are placed side by side, each working its own pump-piston. The piston-rod of each is connected by a short link to a swinging bar, which actuates the slide-valve of the other steam-cylinder. In this way one piston begins its stroke when the motion of the other is about to cease, and a smooth and continuous action is secured[1].

The Worthington engine has been extensively applied, on a large scale, to raise water for the supply of towns and to force oil through "pipe-lines" in the United States. In the larger sizes it is made compound, each high-pressure cylinder having a low-pressure cylinder tandem with it on the same rod. To allow of expansive working, an ingenious device is added which compensates for the inequality of effort on the pump-piston that would result from an early cut-off. A cross-head A (fig. 183) fixed to each of the piston-rods is connected to the plungers of a pair of oscillating cylinders B, B, which contain water and communicate with a reservoir full of air compressed to a pressure of about 300 lb. per square inch. When the stroke (which takes place in the direction of the arrow) begins these plungers are at first forced in, and hence work is at first done by the main piston-rod, through the compensating cylinders B, B,

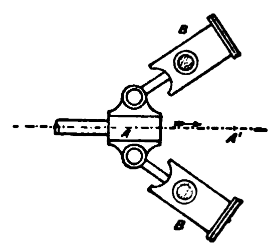


FIG. 183.

[1] For particulars of the performance of several small pumping-engines of this class see *Proc. Inst. Mech. Eng.* Oct. 1893.

on the compressed air in the reservoir. This continues until the crosshead has advanced so that the oscillating cylinders stand at right angles to the line of stroke. Then for the remainder of the stroke their plungers assist in driving the main piston, and the compressed air gives out the energy which it stored in the earlier portion. The volume of the air reservoir is so much greater than the volume of the cylinders *B, B* that the pressure in it remains nearly constant throughout the stroke. Any leakage from the cylinders or reservoir is made good by a small pump which the engine drives. One advantage which this method of equalizing the effort of a steam-engine piston has (as compared with making use of the inertia of the reciprocating masses) is that the effort, when adjusted to be uniform at one speed, remains nearly uniform although the speed be changed, provided the inertia of the reciprocating parts be small. In the Worthington "high-duty" engine, where this plan is in use, the high and low-pressure cylinders are each provided with a separate expansion-valve of the rocking-cylinder type, as well as a slide-valve; the cut-off is early, and the efficiency is as high as in other pumping-engines of the best class. The results obtained in tests of these engines have been referred to in Chapter V.

Another method of compensating for the inequality of the piston thrust during expansion in non-rotative pumping-engines is to connect the pistons not directly but through a rocking piece in such a way that the steam-piston gets a mechanical advantage over the pump-piston as the stroke proceeds. This has been done by Mr Davey in cases where the reciprocating pieces have not enough inertia to make a compensating device unnecessary. A rocking sector between the pistons causes their velocity ratio, which is nearly one of equality in the early portion of the stroke, to alter as the stroke goes on, with the result that in the later stages as the steam pressure falls off the pump-piston moves more slowly than the steam-piston.

236. The Pulsometer. Mr Hall's "pulsometer" is a peculiar pumping-engine without cylinder or piston, which may be regarded as the modern representative of the engine of Savery (§ 6). The sectional view, fig. 184, shows its principal parts. There are two chambers *A, A'*, narrowing towards the top, where the steam-pipe *B* enters. A ball-valve *C* allows steam to pass into one of the

chambers and closes the other. Steam entering (say) the right-hand chamber forces water out of it past the clack-valve V into a delivery passage D, which is connected with an air-vessel. When the water-level in A sinks so far that steam begins to blow through the delivery-passage, the water and steam are disturbed and so brought into intimate contact, the steam in A is condensed, and a partial vacuum is formed. This causes the ball-valve C to rock over and close the top of A, while water rises from the suction-pipe E to fill that chamber. At the same time steam begins to enter the other chamber A', discharging water from it, and the same series of actions is repeated in either chamber alternately. While the water is being driven out there is comparatively little condensation of steam, partly because the shape of the vessel does not promote the formation of eddies, and partly because there is a cushion of air between the steam and the water. Near the top of each chamber is a small air-valve opening inwards, which allows a little air to enter each time a vacuum is formed. When any steam is condensed, the air mixed with it remains on the cold surface and forms a non-conducting layer. Further, when the surface of the water has become hot the heat travels very slowly downwards so long as the surface remains undisturbed. The pulsometer of course cannot claim high efficiency as a thermodynamic engine, but its suitability for situations where other steam-pumps cannot be used, and the extreme simplicity of its working parts, make it valuable in certain cases. Trials of its performance have shown that under favourable conditions a pulsometer may use no more than 150 lbs. of steam per effective horse-power-hour. This consumption does not compare unfavourably with that of small non-rotative steam-pumps[1].

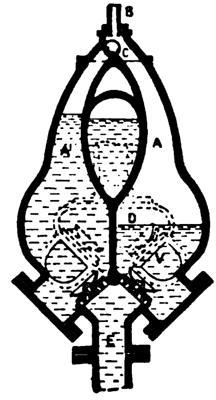


FIG. 184. Pulsometer.

237. Davey's safety Motor. We have seen that the tendency of modern steam practice is towards high pressures, and that this means a gain both in efficiency and in power for a

[1] *Proc. Inst. Mech. Eng.* 1893, p. 456. An automatic valve is described in the same place which enables the pulsometer to use steam expansively.

given weight of engine. High pressure, or indeed any pressure materially above that of the atmosphere, is out of the question when engine and boiler are to work without the regular presence of an attendant. Mr Davey has introduced a small domestic motor which deserves notice from the fact that it employs steam at atmospheric pressure. One form of this engine is shown in fig. 185. The boiler—which serves as the frame of the engine—is of cast-iron in the example shown in the drawing, though in more recent cases a steel boiler has been substituted. It is fitted with a cast-iron internal fire-box, with a vertical flue which is traversed by a water-bridge. The cylinder, which is enclosed within the upper part of the boiler, and the piston are of gun-metal, and work without lubrication. Steam is admitted by an ordinary slide-valve, also of gun-metal, worked by an eccentric in the usual way. The condenser stands behind the boiler; it consists of a number of upright tubes in a box, through which a current of cold water circulates from a supply-pipe at the bottom to an overflow-pipe at the top. In larger sizes of the motor the cylinder stands on a distinct frame, and the boiler has a hopper fire-box, which will take a charge of coke sufficient to drive the engine for several hours without attention. About 6 or 7 lb. of coke are burned per horse-power-hour.

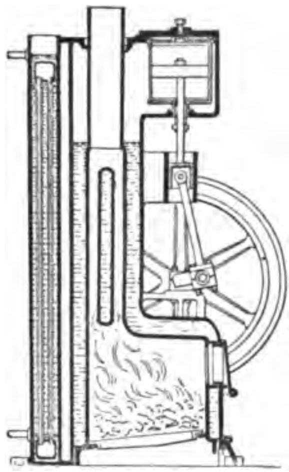


FIG. 185. Davey Motor.

238. Rotary Engines. From the earliest days of the rotative engine attempts have been made to avoid the intermittent reciprocating motion which an ordinary piston-engine first produces and then converts into motion of rotation. The design of *rotary* engines, to use the name generally applied to non-reciprocating forms, has exercised the ingenuity of many inventors, with results which in general have little value or interest except to the student of applied kinematics. Murdoch, the contemporary of Watt, proposed an engine consisting of a pair of spur-wheels gearing with one another in a chamber through which steam passed by

being carried round the outer sides of the wheels in the spaces between successive teeth[1].

In a more modern wheel-engine (Dudgeon's) the steam was admitted by ports in side-plates into the clearance space behind teeth in gear with one another, just after they had passed the line of centres. From that point to the end of the arc of contact the clearance space increased in volume; and it was therefore possible, by stopping the admission of steam at an intermediate point, to work expansively. The difficulty of maintaining steam-tight connexion between the teeth and the side-plates on which the faces of the wheels slide is obvious; and the same difficulty has prevented the success of other forms of rotary engine. These have been devised in immense variety, in many cases, it would seem, with the idea that a distinct mechanical advantage was to be secured by avoiding the reciprocating motion of a piston[2]. In point of fact, however, very few forms entirely escape having pieces with reciprocating motion. In all rotary engines, with the exception of steam turbines,—where work is done by the kinetic impulse of steam,—there are steam chambers which alternately expand and contract in volume, and this action usually takes place through a more or less veiled reciprocation of working parts. So long as engines work at a moderate speed there is little advantage in avoiding reciprocation; the alternate starting and stopping of piston and piston-rod does not affect materially the frictional efficiency, throws no deleterious strain on the joints, and need not disturb the equilibrium of the machine as a whole. The case is different when very high speeds are concerned; it is then desirable as far as possible to limit the amount of reciprocating motion and to reduce the masses that partake in it.

A comparatively recent example of the rotary type in which reciprocating motion occurs only to a trifling extent is the spherical engine of Mr Beauchamp Tower[3]. This engine was, like several of its predecessors[4], based on the kinematic relations of the moving pieces in a Hooke's joint. Imagine a Hooke's joint,

[1] See Farey's *Treatise on the Steam-Engine*, p. 676.

[2] A large number of proposed rotary engines are described, and their kinematic relations to one another are discussed, in Reuleaux's *Kinematics of Machinery*, translated by Prof. Kennedy.

[3] *Proc. Inst. Mech. Eng.*, March 1885.

[4] One of these, the disk-engine of Bishop, was used for a time in the printing-office of *The Times*, but was discarded in 1857.

connecting two shafts set obliquely to one another, to be made up of a central disk to which the two shafts are hinged by semicircular plates, each plate working in a hinge which forms a diameter of the central disk, the two hinges being on opposite sides of the disk and at right angles to one another. Further, let the disk and the hinged pieces be enclosed in a spherical chamber through whose walls the shafts project. As the shafts revolve each of the four spaces bounded by the disk, a hinged piece, and the chamber wall will suffer a periodic increase and diminution of volume, between limits which depend on the angle at which the shafts are set. In Mr Tower's engine this arrangement is modified by using spherical sectors, each nearly a quarter sphere, in place of semicircular plates, for the hinged pieces in which the shafts terminate. The shafts are set at 135° to each other. Each of the four enclosed cavities then alters in volume from zero to a quarter sphere, back to zero, again to a quarter sphere, and again back to zero, in a complete revolution of the shafts. In practice the central disk is a plate of some thickness, whose edge is kept steam-tight in the enclosing chamber by spring-packing, and the sectors are reduced to an extent corresponding to the thickness of the central disk. One shaft is a dummy and runs free, the other is the driving-shaft. Steam is admitted and exhausted by ports in the spherical sectors, whose backs serve as revolving slide-valves. It is admitted to each cavity during the first part of each periodical increase of the cavity's volume. It is then cut off and allowed to expand as the cavity further enlarges, and is exhausted as the cavity contracts. If the working shaft, to which the driven mechanism serves as a fly-wheel, revolves uniformly, the dummy shaft is alternately accelerated and retarded. Apart from this, the only reciprocating motion is the small amount of oscillation which the comparatively light central disk undergoes.

Another rotary engine of the Hooke's-joint family is Mr Fielding's, in which a gimbal-ring and four curved pistons take the place of the disk. Two curved pistons are fixed on each side of the gimbal-ring, and as the shafts revolve these work in a corresponding pair of cavities, which may be called curved cylinders, fixed to each shaft.

239. Steam Turbines. A strictly rotary or non-reciprocating type of engine is found in the steam turbine, where rotation of

a wheel is produced either by impact of a jet upon revolving blades, or by reaction from a jet of escaping steam, as in the æolipile of Hero (§ 2). In order that a revolving piece should extract, either by impulse or reaction, a respectable fraction of the kinetic energy of a steam jet, it must move with immense velocity. The Hon. C. A. Parsons, who has brought the steam turbine to so remarkable a level of efficiency that its performance rivals that of the best piston and cylinder engines, has overcome this difficulty by making the action compound, in other words, he uses a series consisting of several sets of turbine wheels arranged so that the steam acts on each in succession. After leaving the first set of turbine blades the steam passes through a set of fixed guide blades which direct it against the next set of moving blades, and so on, with the result that although only a small part of its energy is taken up by each set, the amount taken up by the whole series is large, and a high efficiency is secured without giving the turbine blades an excessive velocity. A single shaft carries all the turbines. In the original form of the Parsons turbine the wheels were of the central flow type; in a later form the steam flows radially outwards through a series of rings of blades fixed on one face of each wheel. Between each ring and the next is a ring of fixed blades held in position on the face of an annular disk which projects inwards from the circumference of the case in which the turbines are enclosed. After struggling through the series of rings of blades on the first wheel, the steam passes inwards over the back of that wheel to the central region of the next wheel, where again it struggles outwards through alternate rings of revolving turbine blades and fixed guide-blades. The action is repeated on wheel after wheel until the pressure of the steam is reduced to the value at which release is to occur, which may be as low as a good air-pump will produce in a condenser. The turbine is therefore able to work as a condensing steam-engine, and does in fact realise the economic advantage of high expansion. In an example tested by the writer, when the output was at a rate approaching 150 effective horse-power, the steam was expanded from an absolute pressure of 115 lbs. per square inch to 1 lb. per square inch, and the amount of steam which was used per horse-power hour was not more than it would have been in a first-class compound engine.

Fig. 186 shows a portion of this turbine in cross-section

$A_1A_2A_3$ are three of the revolving wheels or discs which carry the turbine blades. $B_1B_2B_3$ are the corresponding fixed rings which carry the guide-blades. The guide-blades project towards the right from the fixed rings B, the turbine blades towards the left

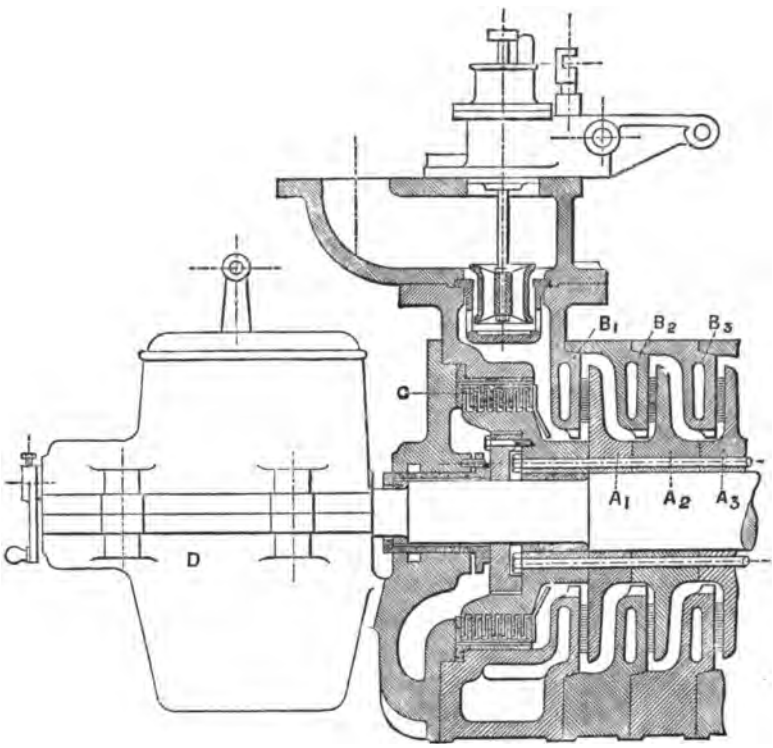


FIG. 186. Section of part of Parsons' Steam Turbine.

from the moving discs A, and fixed and moving blades alike are of such a height that there is very little clearance between their ends and the opposing disc or ring. Steam enters the turbine through a double-beat valve and first reaches the innermost ring of blades between A_1 and B_1. Having acted on the successive rings of blades carried by A_1 it returns towards the centre between the backs of A_1 and B_2, to act in turn on the blades of A_2, then on A_3 and so on. In this example there were six discs A, each 15 inches in diameter, and a seventh, the diameter of which was $26\frac{1}{4}$ inches, and the whole number of rings of turbine blades was 35. The blades are of brass, slightly curved, and set so that the

apertures between them are wider the further the steam has expanded. The one-sided character of the discs would produce an end-thrust on the turbine shaft, but this is balanced by a revolving dummy-piston C, which has a number of deep grooves on its circumference which are entered by corresponding annular projections on a fixed bush resembling the thrust-block of a marine propeller shaft. The high speed of the shaft (4800 revolutions per minute in this instance) requires special forms of bearing and special means of lubrication to be adopted. The main bearings stand in a bath of oil, which is kept in constant circulation by means of a pump, and the bushes in which the shaft turns consist of several concentric sleeves fitting loosely over one another so that a film of oil may find its way between each sleeve and the one outside it. This leaves the shaft some little freedom to adjust itself by lateral displacement, but at the same time the viscosity of the oil films acts as a powerful damper to prevent oscillations from being set up. One of these main bearings is contained in the oil-case D on the left of the figure. The speed is controlled by making the admission of steam take place not continuously, but in gusts at regular intervals, the duration of each gust being automatically regulated (by means of a steam relay governor) to suit the demand for work. The gusts are produced by the periodic lifting and dropping of the double-beat valve shown in the figure. When the engine is working under its full load the gusts become blended into an almost continuous blast.

In the tests referred to above the turbine was used to drive a dynamo, the armature shaft of which was directly coupled to the turbine shaft. Trials were made at various grades of output, ranging up to 137 electrical horse-power, the quantity of steam used per hour being determined in relation to the electrical horse-power developed by the dynamo. The steam was superheated to a moderate extent by passing through pipes in the boiler flue. The air-pump of the condenser was driven by a separate engine, whose consumption of steam is not included in the measurements. The following table gives some of the results, which are also shown in fig. 187, by means of curves drawn in the manner explained in § 145. It will be seen from this figure that the "Willans line" for the steam turbine is very nearly straight, and that the rate of output may be reduced to one-half the full load, or less, with no more than a small increase in the consumption of

steam. If we take the net electrical output to represent say 75 per cent. of the work done by the steam—a proportion which

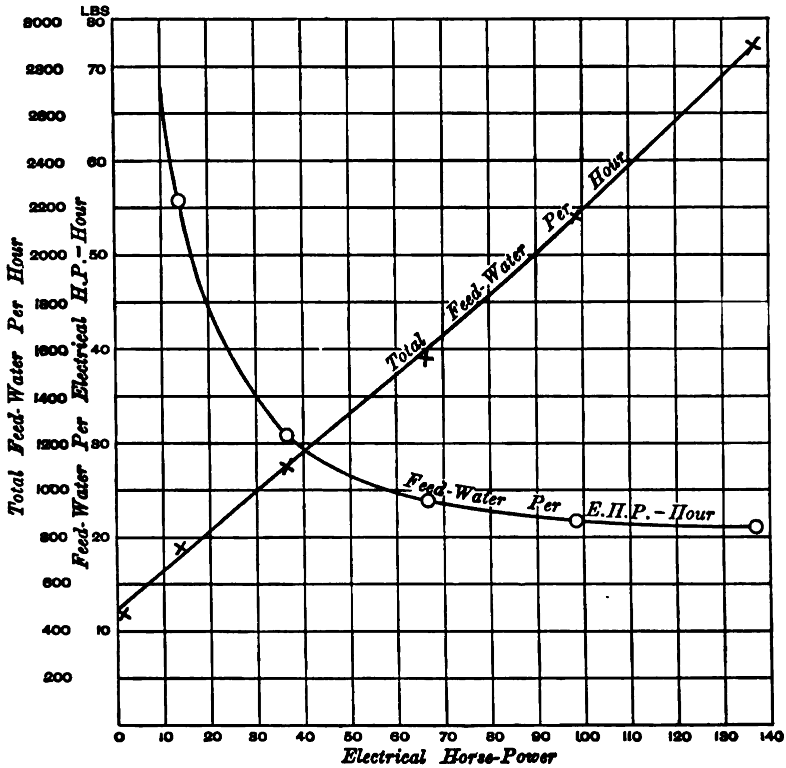


Fig. 187. Results of Steam Turbine Trials.

Table XIII. *Trials of Steam Turbine Dynamo with steam superheated about* 60° F.

Boiler pressure by gauge lbs. per sq. in.	Temperature of steam. ° Fah.	Output in electrical horse-power.	Feed water per hour in lbs.	
			Total	Per electrical h.-p. hour.
96	335	0·13	480	—
102	365	13·7	760	55·6
100	356	36·2	1110	30·7
102	400	66	1590	24·1
100	390	99	2170	21·7
103	398	137	2900	21·2

would roughly hold good in ordinary cases—the performance of the turbine dynamo is equivalent at full load to that of an engine consuming less than 16 lbs. of steam per indicated horse-power-hour. With so moderately high a steam-pressure as 100 lbs. this performance rivals the best examples quoted in Chapter V. The 500 lbs. or so of steam used per hour when the net electrical horse-power is zero is the quantity required to keep the turbine shaft running at its normal speed and to maintain the electrical output of the "exciting" dynamo, which is not included in the figures as part of the net electrical power.

In his most recent turbines Mr Parsons has reverted to the parallel flow type, which allows the construction to be considerably simplified and gives results which are at least equal to those obtained with radial flow. In the parallel flow turbines the revolving portion is a drum with rings of blades projecting from its circumference. The blades are short pieces of rolled or drawn brass caulked into grooves turned on the drum. Between each ring of blades and the next is a corresponding ring of fixed blades which project inwards from the case within which the

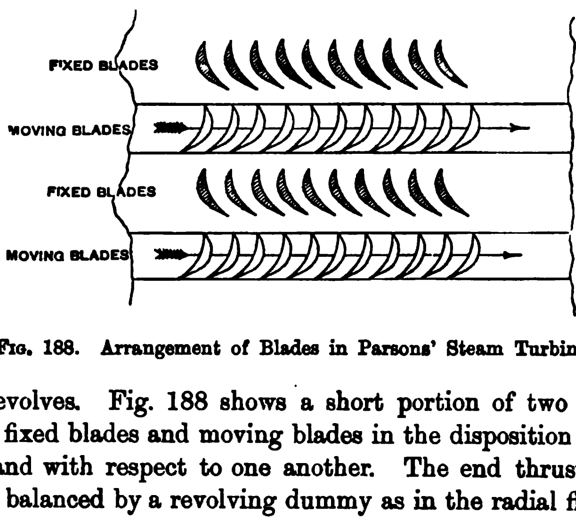


FIG. 188. Arrangement of Blades in Parsons' Steam Turbine.

drum revolves. Fig. 188 shows a short portion of two adjacent rings of fixed blades and moving blades in the disposition in which they stand with respect to one another. The end thrust on the drum is balanced by a revolving dummy as in the radial flow type.

The absence of reciprocating parts gives steam turbines the advantage of complete freedom from vibration. They have been successfully applied not only to the driving of dynamos, ventilating fans, and other high-speed rotative machines, but also

in marine propulsion. The screws of the experimental steamer "Turbinia" (1897) are carried directly on the shafts of three parallel flow turbines, forming a compound series in which steam is expanded from a pressure of 170 lbs. per square inch to about 1 lb. per square inch. Each of the three turbine drums carries a large number of rings of blades through which the steam passes successively. The aggregate horse-power of the combination is about 2100. The turbines weigh much less and take up much less room than ordinary engines of the same power. Trials of the "Turbinia" have shown that a phenomenally high speed was attained with no greater consumption of steam than would have been required to develope the same speed by means of triple-expansion engines of the piston and cylinder type. Estimating the power from the resistance of the vessel, as determined by experiments with a model, it appears that the consumption of steam by the turbines was something less than 14½ lbs. per horse-power-hour.

Another type of steam turbine which has come into considerable use is the jet turbine of De Laval, which resembles a Pelton wheel driven by steam. A jet of steam from a nozzle impinges against a ring of vanes or blades formed on the circumference of a single disc-shaped wheel. The nozzle has a divergent form which causes the steam to expand before it leaves the mouth, and so causes the pressure energy to be used up in giving kinetic energy to the column before it is projected against the vanes. Fig. 189 shows how the jet is made to impinge against the blades on one side, and allowed to escape on the other side after doing its work. The expansion in the nozzle is carried to atmospheric pressure if the steam escapes freely after passing the blades; but by using an ejector condenser on

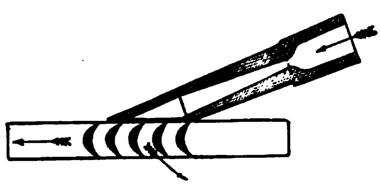


FIG. 189. Jet and Blades in De Laval's Steam Turbine.

the exhaust pipe the back pressure can be brought down to a value much below that of the atmosphere. In the larger sizes of these turbines, as in large Pelton wheels, several jets are used in place of one.

To take full advantage of the kinetic energy of the jet the speed of the blades has to be very high. They are strengthened by shrinking on a steel ring over their tips, which serves also as a cover to prevent the ring of blades from acting as a centrifugal fan. In a 50 horse-power De Laval turbine the shaft makes 15000 revolutions per minute; in the 5 horse-power size it makes as many as 30000 revolutions. To permit of these high speeds the shaft is made flexible, and its speed is geared down by connecting it through helical teeth with a second-motion shaft running at one-tenth of the speed. Trials of a turbine of this type developing (with a condenser) 63 horse-power are reported to have shown an average steam consumption of 19·7 lbs. per B.-H.-P.-hour.

240. Marine Engines. The early steamers were fitted with paddle-wheels, and the engines used to drive them were for the most part modified beam-engines. Bell's "Comet" (§ 21) was driven by a species of inverted beam-engine, and another form of inverted beam, known as the side-lever engine, was for long a favourite with marine engineers. In the side-lever engine the cylinder was vertical, and the piston-rod projected through the top. From a crosshead on the rod a pair of links, one on each side of the cylinder, led down to the ends of a pair of horizontal beams or levers below, which oscillated about a fixed gudgeon at or near the middle of their length. The two levers were joined at their other ends by a crosstail, from which a connecting rod was taken to the crank above. The side-lever engine is now obsolete. In American practice, engines of the beam type, with a braced-beam supported on A frames above the deck, are still found in river-steamers and coasters.

An old form of direct-acting paddle-engine was the steeple-engine, in which the cylinder was set vertically below the crank. Two piston-rods projected through the top of the cylinder, one on each side of the shaft and of the crank. They were united by a crosshead sliding in vertical guides, and from this a return-connecting-rod led to the crank.

Modern paddle-wheel engines are usually of one of the following kinds. (1) In *oscillating cylinder engines* the cylinders are set under the crank-shaft, and the piston-rods are directly connected to the cranks. The cylinders are supported on trunnions which give them the necessary freedom of oscillation to follow the movement of the crank. Steam is admitted through the trunnions to slide-valves on the sides of the cylinders. In some instances the mean position of the cylinders is inclined instead of vertical; and oscillating engines have been arranged with one cylinder before and another behind the shaft, both pistons working on one crank. The oscillating cylinder type is best adapted for what would now be considered comparatively low pressures of steam. (2) *Diagonal engines* are direct-acting engines of the ordinary connecting-rod type, with the cylinders fixed on an inclined bed and the guides sloping up towards the shaft.

When the screw-propeller began to take the place of paddle-wheels in ocean-steamers, the increased speed which it required was at first supplied by using spur-wheel gearing in conjunction with one of the forms of engines then usual in paddle-steamers. After a time types of engine better suited to the screw were introduced, and were driven fast enough to be connected directly to the screw-shaft. The smallness of the horizontal space on either side of the shaft formed an obstacle to the use of horizontal engines, but this difficulty was overcome in several ways. In Penn's trunk-engine, which was at one time a usual form in war vessels, the engine was shortened by attaching the connecting-rod directly to the piston, and using a hollow piston-rod, called a trunk, large enough to allow the connecting-rod to oscillate inside it. The trunk extended through both ends of the cylinder and formed a guide for the piston. The trunk-engine had the drawback of requiring very large stuffing-boxes, of wasting cylinder space, and of presenting a large surface of metal to alternate heating by steam and cooling by contact with the atmosphere.

The return-connecting-rod engine was another horizontal form also used in the navy. It was a steeple-engine placed horizontally, with two, and in some cases four, piston-rods in each cylinder. The piston-rods passed clear of the shaft and the crank, and were joined beyond it in a guided crosshead, from which a connecting-rod returned.

Ordinary horizontal direct-acting engines with a short stroke

and a short connecting-rod have also been used in war-ships, where the horizontal was at one time generally preferred to the vertical type of engine for the sake of keeping the machinery below the water-line. In horizontal marine engines the air-pump and condenser are placed on the opposite side of the shaft from the cylinder, which balances the weight and allows the air-pump to be driven direct.

In merchant ocean-steamers one general type of engine is universal, and the same type is now adopted in naval practice. This is the inverted vertical direct-acting engine. Its most usual form has three cylinders set in line fore and aft above the shaft, working on cranks at 120° from one another, and employing triple expansion. The mechanical advantage of three cranks in giving a nearly uniform turning moment with but little resultant thrust against the shaft has had much to do with the general adoption of this form. A slide-valve serves, without any separate expansion valve, to control the distribution of steam in each cylinder, the cut-off being capable of variation by "notching up" the link-motion or other reversing gear through which the slide-valve is operated. In most instances the cylinders are without steam-jackets.

Surface condensation was introduced in marine practice by S. Hall in 1831, but was not brought into general use until much later. Previous to this it had been necessary, in order to avoid the accumulation of too dense brine in the boiler, to blow off a portion of the brine at short intervals and replace it by sea-water, a process which of course involved much waste of heat. By the use of surface condensers it became possible to use the same feed-water over and over again. The very freedom of the condensed water from dissolved mineral substances was for a time an obstacle to the adoption of surface condensers, for it was found that the boiler, no longer protected by a deposit of scale, became rapidly corroded through the action of acids formed by the decomposition of the lubricating oil. This objection was overcome by introducing a sufficient amount of salt water to allow some scale to form, and the use of surface condensers soon became universal on steamers plying in sea-water. The marine condenser consists of a multitude of tubes, generally of brass, about $\frac{3}{4}$ of an inch in diameter. Through these cold sea-water is made to circulate, while the steam is brought into contact with their outside surfaces.

In some cases, especially in Admiralty practice, cold water circulates outside the tubes and the steam passes inside.

The ordinary marine engine has four pumps:—the air-pump, which is made large enough to serve in case injection instead of surface-condensation should at any time be resorted to; the feed-pump; the circulating-pump, which maintains a current of sea-water through the tubes of the condenser; and the bilge-pump, which discharges any water that may gather by leakage or otherwise in the bilge of the ship. The pumps are so arranged that in the event of a serious leak the circulating-pump can also draw its supply from the bilge. In many engines, especially those of less recent construction, the four pumps are placed behind the condenser, and are worked by a single crosshead driven by a lever, the other end of which is connected by a short link with one of the crossheads of the engine. It is now usual to have a small engine, distinct from the main engine, to drive the feed-pump, and the circulating water is often supplied by a centrifugal or other pump also driven by a separate engine[1].

241. Relation of power to weight in Marine Engines. In the improvement of the marine engine two points have been particularly aimed at,—reduction in the rate of consumption of coal per horse-power, and reduction in the weight of the machine (comprising the engine proper and the boiler) per horse-power. The second consideration is in some cases of even more moment than the first, especially in war-ships. Progress has been made, in both respects, by increase of steam-pressure, and, in the second respect especially, by increase of piston speed. The gain in economy which came with rise in boiler pressure has been referred to in § 21. As to the reduction in weight, Messrs Marshall and Weighton, in a paper dealing with this subject[2], have pointed out that before the introduction of triple expansion and forced draught the weight of engines in the mercantile marine, including the

[1] On the general subject of marine engines, reference should be made to Mr A. E. Seaton's *Manual of Marine Engineering*; to Mr R. Sennett's *Treatise on the Marine Steam-Engine*; and to Mr W. H. Maw's *Recent Practice in Marine Engineering*. For particulars of the engines and boilers of modern war-vessels see the paper by Sir A. J. Durston referred to in § 21.

[2] Marshall and Weighton, *Proc. North-East Coast Inst. Engineers and Ship-builders*, 1886.

boilers and the water in them, was 480 lb. per I.-H.-P. In the navy this was reduced, chiefly by the use of lighter framing with the object of minimizing weight, to 360 lb. Triple engines of the merchant type, without forced draught, are only slightly lighter than double engines; but in naval practice, where forced draught, greatly increased speed, and the use of steel for frames and working parts have combined to reduce the ratio of weight to power, a marked reduction in weight is apparent. To quote examples, a set of vertical triple engines, which indicate 2200 H.-P. with natural draught, and 4000 H.-P. with a draught forced by pressure in the stokehole equal to 2 inches of water, weigh under the latter condition (along with the boilers) only 155 lb. per I.-H.-P. In another set, in which the draught is forced by a pressure of 3 inches, and the cylinders are only 15½, 24 and 37 inches in diameter, with a stroke of 16 inches, the indicated horse-power is 4200, and the weight of engines and boilers is 136 lb. per I.-H.-P. In these the boilers are of the locomotive type, and the mean piston speed is 1066 feet per minute. Even these light weights are surpassed in smaller engines, such as those of torpedo-boats. In so far as this large development of power from a small weight of machinery is due to high piston speed, it is secured without loss—indeed with some gain—of thermodynamic efficiency; forced draught, however, without a corresponding extension of the heating surface, leads to a less efficient expenditure of fuel. With a given type of engine there is a certain ratio of expansion which gives a minimum in the ratio of weight to power; when this ratio of expansion is exceeded the engines have to be enlarged to an extent that more than counterbalances the saving in boiler weight; when a less ratio of expansion is used the boilers have to be enlarged to an extent that more than counterbalances the reduction of weight in the engine proper.

The application of the steam turbine to marine propulsion makes it practicable to reduce the weight of the motor to an extent much exceeding any reduction that has been achieved with piston and cylinder engines. The turbine engines of the "Turbinia," which develope 2100 horse-power, weigh (when taken alone) less than four tons, and the total weight of the machinery on that vessel, including along with the engines the boiler, condenser, water-tanks and all auxiliary mechanism, is 22 tons, which is only 24 lbs. per horse-power. The exceptional ratio of

power to weight in this instance is due partly to the lightness of the turbines as compared with ordinary engines and partly to the use of a severe forced draught with a water-tube boiler. The draught was produced by a fan on one of the turbine shafts blowing air into the stokehole under a pressure of about 8 inches by water-gauge.

242. Locomotives. The ordinary locomotive consists of a pair of direct-acting horizontal or nearly horizontal engines, fixed in a rigid frame under the front end of a boiler of the type described in § 221, and coupled to the same shaft by cranks at right angles, each with a single slide-valve worked by a link-motion, or by a form of radial gear. The engine is non-condensing, except in special cases, and the exhaust steam, delivered at the base of the funnel through a blast-pipe, serves to produce a draught of air through the furnace. In some instances a portion of the exhaust steam, amounting to about one-fifth of the whole, is diverted to heat the feed-water. In tank engines the feed-water is carried in tanks on the engine itself; in other engines it is carried behind in a tender.

On the shaft are a pair of driving-wheels, whose frictional adhesion to the rails furnishes the necessary tractive force. In some engines there is a single pair of driving-wheels; in many more a greater tractive force is secured by having two equal driving-wheels on each side, connected by a coupling-rod between pins on the outside of the wheels. In goods engines a still greater proportion of the whole weight is utilized to give tractive force by coupling three and even four wheels on each side. These arrangements are distinguished by the terms "four-coupled," "six-coupled," and "eight-coupled" applied to the engines. In *inside-cylinder* engines the cylinders are placed side by side within the frame of the engine, and their connecting-rods work on cranks in the driving shaft. In *outside-cylinder* engines the cylinders are spread apart far enough to lie outside the frame of the engine, and to work on crank-pins on the outsides of the driving-wheels. This dispenses with the cranked axle, which is apt to be the weakest part of a locomotive engine. Owing to the frequent alternations of strain to which it is subject, a locomotive crank-axle is peculiarly liable to rupture, and has to be removed after a certain amount of use.

In some locomotives the leading wheels are coupled to driving-wheels behind them, but it is now generally preferred to have under the front of the engine two or four smaller wheels which do not form part of the driving system. These are carried in a *bogie*, that is, a small truck upon which the front end of the frame rests by a swivel-pin or plate which allows the bogie to turn, so as to adapt itself to curves in the line, and thus obviate the grinding of tyres and danger of derailment which would be caused by using a long rigid wheel-base. The bogie appears to have been of English origin[1]; it was brought into general use in America, and is now common in English as well as in American practice. Instead of a four-wheeled bogie, a single pair of leading wheels are also used, carried by a Bissel *pony* truck, which has a swing-bolster pivoted by a radius bar about a point some distance behind the axis of the wheels. This has the advantage of combining lateral with radial movement of the wheels, both being required if the wheel-base is to be properly accommodated to the curve. Another method of getting lateral and radial freedom is the plan used by Mr Webb of carrying the leading axle in a box curved to the arc of a circle, and free to slide laterally for a short distance, under the control of springs, in curved guides[2].

In inside-cylinder engines the slide-valves are frequently placed back to back in a single valve-chest between the cylinders. The width of the engine within the frame leaves little room for them there, and they are reduced to the flattest possible form, in some cases with split ports, half above and half below a partition in a central horizontal plane. In a few engines the valves are below the cylinders, with faces sloping down towards the front, while the cylinders themselves slope slightly up. In many more the valves work on horizontal planes above the cylinders; this position is specially suitable when Joy's or some other form of radial gear is used instead of the link-motion. Radial valve-gears have the advantage, which is of considerable moment in inside-cylinder engines, that the part of the crank-shaft's length which would otherwise be needed for eccentrics is available to increase the width of main bearings and crank-pins, and to strengthen the crank-cheeks. Walshaert's gear is very extensively used on Continental locomotives, and Joy's has been applied to a large number of British engines.

[1] *Min. Proc. Inst. C. E.*, vol. liii. p. 50. [2] *Proc. Inst. Mech. Eng.*, 1883.

The outside-cylinder type is adopted by several British makers; in America it is universal. There the two castings which form the cylinders are bolted together to make a saddle on which the bottom of the smoke-box sits. The slide-valves are on the tops of the cylinders, and are worked through rocking levers from an ordinary link-motion.

243. Compound Locomotives. Locomotive engines have been compounded in several ways; but it is still an open question whether the application of compound working to locomotives offers any distinct advantage, when regard is had to convenience in driving and cost of repairs as well as to economy in fuel.

In 1876 Mr A. Mallet introduced, on the Bayonne and Biarritz Railway, a type of compound locomotive in which one small high-pressure cylinder and one large low-pressure cylinder were used in place of the two equal cylinders of a common locomotive. Outside cylinders were used in the first instance, but Mallet's system is also applied to inside-cylinder engines. The pipe from the high to the low-pressure cylinder takes a winding course through the smoke-box; this gives it a sufficient capacity to serve as intermediate receiver, and also dries the steam before it enters the large cylinder. A reducing valve is provided through which steam of a pressure lower than that of the boiler can be admitted direct to the low-pressure cylinder to facilitate starting. The reversing gear is arranged to act on both cylinders by one movement, and also to permit a separate adjustment of the cut-off in each. Engines on Mallet's system have been successfully used on other Continental railways and in India, in some instances by conversion from the non-compound form. His plan has the advantage of permitting this conversion to be made (in certain cases), and of requiring scarcely any more working parts than are needed in a common locomotive; but it gives an unsymmetrical engine. He has also proposed an engine with four cylinders,—one high-pressure cylinder tandem with one low-pressure cylinder on each side. Another symmetrical form has been used, in which a pair of outside high-pressure cylinders are compounded with a pair of inside low-pressure cylinders. In England, on the North-Eastern Railway, Mr T. W. Worsdell has made extensive use of compound engines, employing two cylinders only which stand side by side inside the frame with valves on the top worked by Joy's gear. In America the Baldwin company use a pair of

compound cylinders on each side of the engine; the cylinders are set one directly above the other, and their volume ratio is three to one.

The most important experiment yet made in this direction is that which Mr F. W. Webb, of the London and North-Western Railway, has been conducting on a large scale since 1881. In Mr Webb's system three cylinders are used. Two equal high-pressure cylinders are fixed outside the frames, and drive the rear driving axle by crank-pins set at right angles to one another. A single low-pressure cylinder of very large size is placed beneath the smoke-box, and drives a crank in the middle of the forward driving axle. The driving axles are not coupled, and the phase-relation of the low-pressure to the high-pressure stroke is liable to alter through unequal slip on the part of the wheels. This, however, is of no material consequence, on account of the large size of the intermediate receiver and the uniformity with which the two high-pressure cylinders deliver steam to it. The receiver is formed, as in Mr Mallet's arrangement, by leading long connecting pipes through the smoke-box. All three slide-valves are worked by Joy's gear. Those of the low-pressure cylinders are placed below the cylinders (an arrangement which has the advantage of letting the valve fall away from the port-face when the engine is running down-hill with the steam-valve closed); the valve of the large cylinder is above it. The design is completely symmetrical; it has the important mechanical advantage of dispensing with coupling rods, while retaining the greater tractive power of four drivers; only one axle is cranked, and that with a single crank in the centre, which leaves ample room for long bearings.

244. Tramway and Road Locomotives. Tramway locomotives for the most part resemble railway locomotives in the general features of their design. The boiler is of the usual locomotive type. A pair of cylinders in front, either inside or outside the frames, are connected directly to the hindmost of two coupled driving axles. Owing to the smallness of the driving-wheels the axles lie near the road, and the cylinders are set sloping at a considerable angle upwards away from the cranks to keep them clear of dirt. To prevent the discharge of steam into the atmosphere, the exhaust steam is often led into an atmospheric

condenser, consisting of a large number of pipes placed on the top of the engine, and exposed to free contact with the air. In some instances the common locomotive type is widely departed from: a mixed vertical and horizontal boiler is used, and the engine is connected to the driving axle by worm-wheel or other gear, or by a rocking lever between the connecting-rod and the crank[1].

In the "fireless" tramway locomotive of Mr Léon Francq, a reservoir which takes the place of an ordinary boiler is charged at the beginning of the journey with water heated under pressure by injecting steam from stationary boilers at a pressure of 15 atmospheres. The thermal capacity of the water is sufficient —without further addition of heat—to supply steam to the engine during the journey, at a pressure which gradually falls off[2]. The system has not come into general use.

Several forms of tramway engine have been devised in which the motive power is supplied by compressed air[3]. In the Mekarski system the compressed air, on its way from the reservoir to the cylinders, passes through a vessel containing hot water and steam under pressure (charged, as in Francq's system, by injecting steam at a station). In this way the air is heated, and may then expand in the cylinder without having its temperature lowered to an objectionable degree.

Steam road-locomotives or traction-engines have usually a boiler of the locomotive type, with a cylinder or compound pair of cylinders, generally on the top, driving a shaft from which motion is taken by a gearing chain or spur-wheels to a single driving axle at the fire-box end. The engine is steered by means of a leading axle, whose direction is controlled by a hand-wheel and chain-gear. To facilitate rapid turning the driving-wheels are connected to their axle by a differential or compensating gear which allows them to revolve at different speeds. This is a set of four bevel-wheels like White's dynamometer coupling: the outside bevel-wheels are attached to the driving-wheels; the intermediate ones, which gear with these, turn in bearings in a revolving wheel driven by the engine. So long as both driving-wheels are equally resisted both are driven at the same speed, but if one is retarded (as the inner wheel is in going round a curve) it

[1] See *Min. Proc. Inst. C. E.*, vol. xxix., 1884; also *Proc. Inst. Mech. Eng.*, 1880.

[2] *Proc. Inst. Mech. Eng.*, 1879.

[3] *Proc. Inst. Mech. Eng.*, 1878, 1881.

acts to some extent as a fulcrum to the bevel gear, and the other wheel takes a greater share of the motion.

An important feature in traction-engines is the elasticity of the driving-wheels. Many devices have been employed, partly to give the wheels an extended tread, or arc of contact with the ground, and partly to avoid shocks in passing over rough ground[1]. In some designs the rims have been made elastic, as in Mr R. W. Thomson's road steamer, where each wheel had a thick tyre of india-rubber, protected on the outside by an armour of small plates. The earliest pneumatic tyre was invented by Mr Thomson in 1845 for use on vehicles of this class. In other examples the spokes have had the form of springs allowing the axle to take an eccentric position. In others still the framework of the wheel is nearly rigid, but the circumference is filled with blocks of wood which are held in cells with an elastic pad behind each, so that each block in turn is pushed in when it becomes part of the surface over which the weight is distributed, and the tread of the wheel is consequently enlarged.

The Serpollet steam carriage is an interesting form which has had considerable vogue in France for use both on roads and on tramways. Its distinctive feature is the steam generator, which consists of a group of comparatively thick steel tubes, partially flattened to reduce the bore. The furnace keeps these at a high temperature, and steam is formed as it is wanted by pumping in a small quantity of water at a time, which is immediately evaporated and superheated by taking up heat from the substance of the tubes. The generation of steam is governed by controlling the feed. When the feed is too rapid steam accumulates in the boiler and raises the pressure beyond its normal limit. This causes a by-pass to open which diverts a portion of the feed-water from entering the boiler.

[1] *Min. Proc. Inst. C. E.*, vol. xxxvi., 1873; vol. ciii., 1890.

CHAPTER XIII.

AIR-ENGINES.

245. Air-engines with external or internal combustion. The term Air-engine may be used in a restricted sense to denote an engine in which the working substance is atmospheric air, but it may with advantage be extended to apply to any heat-engine which employs a gaseous working substance, as distinguished from a vapour which becomes condensed during some part of the cycle of operations. In this more extended sense the term would include engines in which the working substance is the mixed gas resulting from the combustion of fuel, whether gaseous, liquid or solid, within the engine itself. In other words, it would include gas-engines and oil-engines. These will be more particularly considered in the next chapter, but some of the remarks which follow, relating to air-engines in general, apply to them as well as to engines using atmospheric air.

When air alone forms the working substance, it receives heat from an external furnace by conduction through the walls of a containing vessel, just as the working substance in the steam-engine takes in heat through the shell of the boiler. An engine supplied with heat in this way may be called an *external-combustion* engine, to distinguish it from the very important class of engines in which the combustion which supplies heat occurs within a closed chamber containing the working substance. The ordinary coal-gas explosive engine is the most common type of *internal-combustion* engine.

Compared with engines using saturated steam, engines using air or other gases have the advantage that the temperature and the pressure of the working substance are independent of one

another. In the steam-engine, and in any other heat-engine in which the working substance is a saturated vapour, the upper limit of temperature is comparatively low in consequence of the high pressure with which high temperature is, in such cases, necessarily associated. But in an air or gas-engine it becomes possible to use an upper limit of temperature greatly higher than the limit in the ordinary steam-engine, and if the lower limit is not correspondingly raised an increase of thermodynamic efficiency results. It is true that the upper limit of temperature may be raised in the case of steam, by superheating; but even when the amount of superheating is exceptionally great a steam-engine continues to take in the greater part of its supply of heat at the comparatively low temperature at which the feed-water is converted into steam, and the direct thermodynamic advantage is consequently small.

So long as external combustion is used, there must still be some considerable drop in temperature, of an irreversible and therefore wasteful kind, between the temperature which is produced by combustion in the furnace, and the temperature at which the working air receives its heat, since without this no sufficiently rapid conduction of heat through the walls of the heater could occur. Internal-combustion engines have the advantage that the temperature which is produced in the combustion is itself the upper limit in the thermodynamic cycle.

246. Air-engine using Carnot's cycle. A simple, thermodynamically perfect form of external-combustion air-engine would be one following Carnot's cycle, in which heat is received while the air is at the highest temperature τ_1, the air meanwhile expanding isothermally. After this the supply of heat is stopped, and the air is allowed to expand adiabatically until its temperature falls to the lower extreme τ_2. At this it is compressed isothermally, giving out heat, and finally the cycle is completed by adiabatic compression, which restores the initial high temperature τ_1. The indicator diagram for this cycle has been sketched in fig. 12, § 41. Practically, this action would be attended by the serious drawback that the volume to be swept through by the piston would be very great in relation to the work done. The inclination of adiabatic to isothermal curves for a gas is slight, and hence the area of the diagram, or the effective work done per revolution, is small in comparison with the two quantities of which

it is the difference, namely, the work done by the substance during the forward stroke and the work spent upon it during the backward stroke. An air-engine using Carnot's cycle would consequently be excessively bulky and mechanically inefficient.

247. External Combustion Air-engine with Regenerator: Stirling's Air-engine. This objection is much lessened when the use of a regenerator (§ 51) is substituted for the adiabatic steps of the Carnot cycle. In Stirling's engine, where the regenerator was first used, the working substance was cooled from the upper limit τ_1 to the lower limit τ_2 by passing in one direction through a regenerator, which stored the heat it extracted from the gas in such a way that when the gas was passed through the regenerator in the opposite direction the heat was again taken up and the temperature consequently rose from τ_2 to τ_1. The cycle of operations has been described in § 52, and an ideal indicator diagram has been sketched in fig. 13.

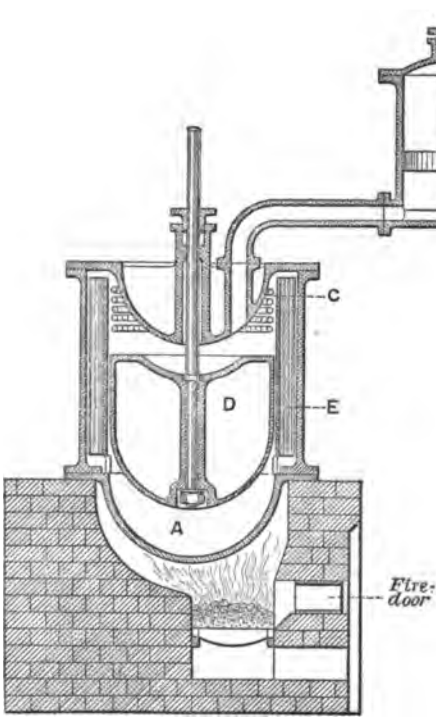


FIG. 190. Stirling's Air-engine

Several forms of engines were designed by Stirling in which the action approximated to the cycle of § 52. The characteristic parts of one of them are shown in section in fig. 190. *A* is the heater, a closed iron vessel containing air, externally heated by a furnace beneath it. A pipe from the top of *A* leads to the working cylinder *B*. At the top of *A* is a cooler *C*, consisting of pipes through which cold water circulates. In *A* there is a displacer plunger *D*, which is driven by the engine; when this is raised the air in *A* is in the lower part of the vessel and is consequently taking in heat from the furnace, whereas when *D* is lowered the air in *A* is transferred from the lower to the upper part and is thereby brought into contact with the refrigerator. On its way from the bottom to the top of *A*, or from the top to the bottom, the air must pass through an annular lining of wire-gauze *E*. This is the regenerator, and the air in passing up through it becomes cooled, and in passing down again through it becomes heated. At the beginning of the cycle *D* is at or near its highest position. The air is then receiving heat at temperature τ_1, and is expanding isothermally; this is the first stage in § 52. Then the plunger *D* descends. The air is driven through the regenerator, where it deposits heat, and its temperature on emerging at the top is τ_2. Next, the working-piston makes its down-stroke (in the actual engine the working cylinder was double-acting, another heating vessel, precisely like *A*, being connected with the cylinder *B* above the piston); this compresses the air isothermally, the heat produced by compression being taken up by the cooler *C*. Finally the plunger is raised, and the working air again passes down through the regenerator, taking up the heat it left there, and rising in temperature to τ_1.

The actual forms in which Stirling's engine was used are described in two patents by R. & J. Stirling (1827 and 1840[1]). An important feature in them was that the air was compressed by means of a pump which formed an additional organ of the engine, so that its average pressure was kept much above that of the atmosphere. Stirling's cycle is theoretically perfect whatever be the density of the working air, and the use of compression does not increase what may be called the theoretical thermodynamic

[1] The 1827 patent is reproduced in Fleeming Jenkin's Lecture on Gas and Caloric Engines, *Inst. Civ. Eng.*, Heat Lectures, 1883-84. See also *Min. Proc. Inst. C. E.*, 1845 and 1854.

efficiency. It does, however, very greatly increase the mechanical efficiency, and also, what is of special importance, it increases the amount of power developed by an engine of given size. To see this it is sufficient to consider that with compressed air a greater amount of heat is dealt with in each stroke of the engine, and therefore a greater amount of work is done. Practically the use of compressed air also increases the thermodynamic efficiency by reducing the ratio of the heat wasted by external conduction and radiation to the whole heat.

A double-acting Stirling engine of 50 I.-H.-P., used in 1843 at the Dundee foundry, appears to have realized an efficiency of 0·3, and, notwithstanding very inadequate means of heating the air it consumed only 1·7 lb. of coal per I.-H.-P.-hour[1]. This engine remained at work for three years, but was finally abandoned on account of the failure of the heating vessels. In one form of the engine as described in Stirling's patent the regenerator was a separate vessel; in another the plunger D was itself constructed to serve as regenerator by filling it with wire-gauze and leaving holes at top and bottom for the passage of the air through it.

248. Ericsson's Air-engine. Another mode of using the regenerator was introduced in America by Ericsson, in an engine which also failed, partly because the heating surfaces became burnt, and partly because their area was insufficient. In Ericsson's engine, which was tried on a considerable scale on the steam-ship "Caloric," the temperature of the working substance was changed by passing through the regenerator while the pressure remained constant. Cold air was compressed by a pump into a receiver, from which it passed through a regenerator into the working cylinder. In so passing it absorbed heat from the regenerator and expanded. The air in the cylinder was then allowed to expand further by taking in heat from a furnace under the cylinder until its pressure fell to near that of the atmosphere. The cycle was completed by the discharge of the air through the regenerator. The indicator diagram approximates to a form bounded by two isothermals and two lines of constant pressure[2].

[1] See Rankine's *Steam-Engine*, p. 367. The consumption per brake H.-P. was much greater.

[2] For a diagram of Ericsson's engine see Rankine's *Steam-Engine*, or *Proc. Inst. Mech. Eng.*, 1873.

249. Modern Air-engines of the Stirling type. Externally-heated air-engines are now employed only for very small powers—from a fraction of 1 H.-P. up to about 3 H.-P. Powerful engines of this type are scarcely practicable, partly on account of the relatively enormous bulk they would have and partly on account of the difficulty which would be experienced in the heating of large quantities of air. By keeping the working substance highly compressed, giving it a mean density much in excess of that of atmospheric air, the bulk of the working cylinder and displacer might be reduced, but the difficulty would remain of getting enough heating surface and of preserving the heater from being burnt through its exposure to oxygen at a high temperature. The small engines of this type that are now manufactured resemble the original Stirling engine very closely in the main features of their action, and comprise essentially the same organs.

One of these modern Stirling engines is the small domestic

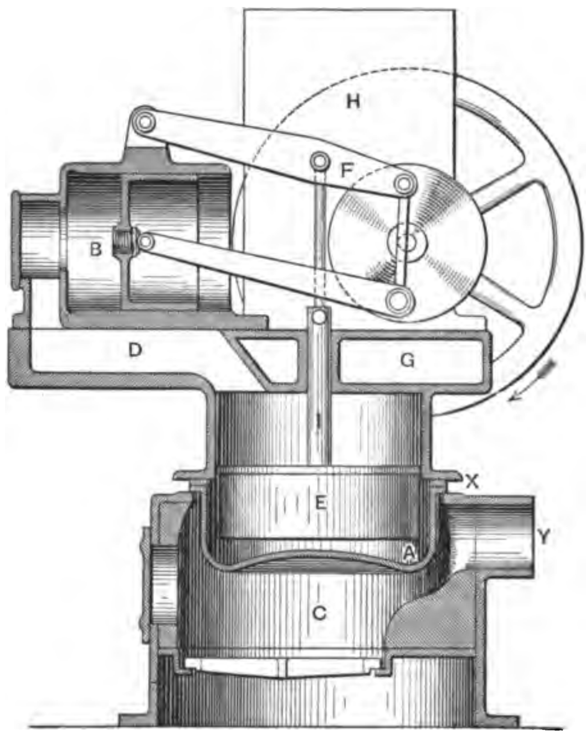


FIG. 191. Robinson's form of Stirling Engine.

motor manufactured under the patents of Mr H. Robinson (fig. 191). In this case there is no compressing pump and the mean pressure of the working air is equal to the pressure of the atmosphere. The range of pressure is slight—so slight indeed that no packing is needed in the piston or other working parts—and the engine developes only a fraction of one horse-power.

A is the heater and displacer cylinder; *B* is the working cylinder, which communicates with *A* by a passage *D*. *A* is heated externally by a small coke fire at *C* or by a gas flame from a Bunsen burner. The displacer *E* takes its motion from a rocking lever *F* connected by a short link to the crank-pin, and is about 90° in phase ahead of the working piston. In the figure the displacer is at the bottom of its stroke and the piston has still half the back stroke to perform. The displacer *E* is itself the regenerator, its construction being such that the air passes up and down through it as in one of the original Stirling forms. On the top of the displacer cylinder is a water vessel *G*, which is the cooler, and this is kept in communication with the circulating water tank *H*. The account which has already been given of the Stirling cycle will serve as a description of the action in this engine. A conspicuous feature is that there are neither valves, packing, nor glands; but the absence of compression, which makes this possible, limits the efficiency of the engine as well as its power.

A larger engine of the Stirling type, working up to some 3 horse-power, is made by Messrs Bailey, of Salford. Another, the Rider engine, made by Messrs Hayward and Tyler, follows substantially but not exactly the Stirling cycle. A sectional view of this engine is given in fig. 192. *A* and *B* are two cylinders, open at the top, with plunger pistons *C* and *D*, which are connected to cranks nearly at right angles. Between the two is the regenerator *H*. Round the lower part of *C* is the cooler *E*, a jacket through which cold water is kept circulating. Under the lower part of *B* is the furnace *F* which heats the air contained in the space *G* below the plunger *D*. In the position shown in the figure, *D* is rising, and *C* is just beginning to rise. Nearly all the working air has been compressed into *G* and is expanding as it takes in heat, doing work against the plunger *D* and also against *C*. By the time *D* reaches the top of its stroke *C* is about half way up: air is passing rapidly through the regenerator from *G* to the space under *C*, and is cooled first by the regenerator and then

by the water-jacket *E*. As *D* comes down this transfer of the air continues and the pressure falls. Then *C* follows, compressing

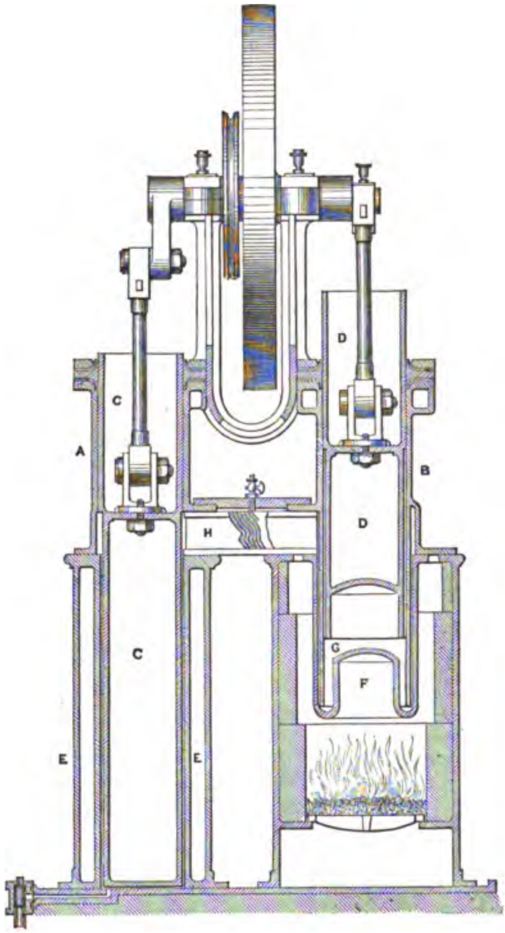


FIG. 192. Rider's Hot Air Engine.

the air beneath it while the cooler *E* absorbs the heat, and finally forcing all the working air back through the regenerator into the heater, when the cycle begins again. The maximum pressure reached during the cycle is about 20 lbs. per sq. inch.

The action is of course continuous, but we may broadly distinguish the following four stages:

(1) The air, previously compressed to a small volume (in *G*),

takes in heat at its highest temperature and expands, doing work on D and subsequently to some extent on C.

(2) After this expansion it is transferred through the regenerator to the cold cylinder A, storing heat in the regenerator and losing pressure. During this process little work is done on or by the air, since the actions on the plungers nearly balance. In other words, the volume does not materially change.

(3) The air which is now in A, expanded to large bulk and at a low temperature, is compressed by the descent of C and gives out heat to the cooler E. During this process work is done upon the air by the fly-wheel.

(4) The compressed air is transferred through the regenerator to G, rising in temperature and pressure. In this process, again, little work is done by or on the air.

This engine differs from the pure Stirling type chiefly in having a displacer which is also a working piston. The Rider engine is mainly used to pump water for domestic supply, and the cooling jacket E is kept cold by making the water which the engine pumps circulate through it.

250. Internal Combustion Air-engines. The difficulty already referred to of getting heat into and out of a gaseous working substance is a fundamental objection which has prevented the external-combustion air-engine from coming into use on any large scale. The activity of a heating surface is vastly greater when the substance that is being heated is changing its state from liquid to vapour than when the substance is already a gas. And similarly a gas that is being cooled by conduction through a surface will part with its heat far less readily than a vapour which is being condensed in the process.

This objection applies when the air-engine is of the external-combustion type, but so far as the heating process is concerned it is removed by causing the combustion to occur within the engine itself. So far as cooling is concerned the difficulty also disappears when the substance which is to reject heat is expelled to the atmosphere instead of being used over again after cooling. Hence it has been possible for the internal combustion engine to attain a much greater efficiency.

The earliest practical example of the internal combustion engine (if we leave guns out of account) appears to have been the

hot-air engine of Sir George Cayley[1], of which Wenham's[2] and Buckett's[3] engines are recent forms. In these engines coal or coke is burnt under pressure in a closed chamber, to which the fuel is fed through a species of air-lock. Air for combustion is supplied by a compressing pump, and the engine is governed by means of a distributing valve which supplies a greater or less proportion of the air below the fire as the engine runs slow or fast. The products of combustion, whose volume is increased by their rise in temperature, pass into a working cylinder, raising the piston. When a certain fraction of the stroke is over the supply of hot gas is stopped, and the gases in the cylinder expand, doing more work and becoming reduced in temperature. During

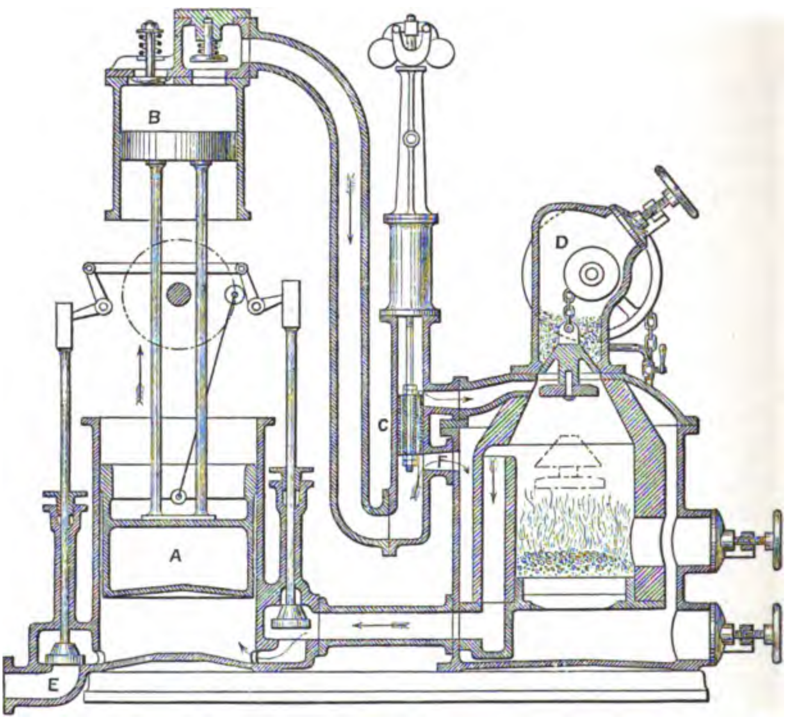


FIG. 193. Buckett's Internal Combustion Air-engine.

the return stroke they are discharged into the atmosphere, and the pump takes in a fresh supply of air. Fig. 193 is a diagram

[1] *Nicholson's Art Journal*, 1807. See also *Min. Proc. Inst. C. E.*, vol. ix.
[2] *Proc. Inst. Mech. Eng.*, 1873. [3] Fleeming Jenkin, *loc. cit.*

section of the Buckett engine. A is the working piston, the form of which is such as to protect the tight sliding surface (at the top) from contact with the hot gases; B is the compressing pump, and C is the valve by which the governor regulates the rate at which fuel is consumed by admitting more or less of the air under the grate through the channel F. D is the air-lock and hopper through which fuel is supplied, and E is the exhaust valve through which the products of combustion are finally expelled.

In engines of this class the degree to which the action is thermodynamically efficient depends very largely on the amount of cooling the gases undergo by adiabatic or nearly adiabatic expansion under the working piston. Without a large ratio of expansion the thermodynamic advantage of a high initial temperature is lost. In any kind of internal-combustion engine the gases have to be discharged at atmospheric pressure, and consequently a large ratio of expansion is possible only when there is much initial compression. Compression is therefore an essential condition, without which a heat-engine of this type cannot be made efficient. It is also, as has already been pointed out, an essential feature in any air-engine which is to develope a fair amount of power without excessive bulk.

Internal-combustion engines using solid fuel have hitherto been but little used, and that only for small powers. But the use of liquid and especially of gaseous fuel has given the internal-combustion engine a position of great and constantly increasing importance. Gas-engines, that is to say, engines acting by the combustion or explosion of a mixture of air and a combustible gas have in recent years entered into serious competition with the steam-engine. In most of these the fuel is ordinary coal-gas; it may however be a cheaper combustible gas, such as that produced by Mr Dowson's process; and in a number of modern examples of the internal-combustion engine the fuel is petroleum, generally vaporised before its admission to the cylinder in which combustion is to occur. Some of these forms will be considered more particularly in the next chapter.

CHAPTER XIV.

GAS-ENGINES AND OIL-ENGINES.

251. Early Gas-engines. Lenoir. The first gas-engine to be brought into practical use was that of Lenoir (1860)[1]. During the early part of the stroke air and gas, in proportions suitable for combustion, were drawn into the cylinder. At about half-stroke the inlet valve closed, and the mixture was immediately exploded by an electric spark. The heated products of combustion then did work on the piston by expanding during the remainder of the forward stroke, and were expelled during the back stroke. The engine was double-acting, and the cylinder was prevented from becoming excessively heated by a casing through which water was kept circulating. This water-jacket is a feature that has been retained in nearly all later gas-engines. In Lenoir's engine every stroke was active, two explosions taking place per revolution, one on each side of the piston.

An indicator diagram from a Lenoir engine is shown in fig. 194[2]. After the explosion the line falls, partly from expansion, and partly from the cooling action of the cylinder walls; on the other hand, its level is to some extent maintained by the phenomenon of after-burning, which will be discussed later. In this engine, chiefly because there was no compression of the explosive mixture before it was ignited, the heat removed by the water-jacket bore an

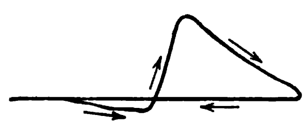


FIG. 194. Lenoir Engine Diagram.

[1] For a full account of the early history of the gas-engine as well as for descriptions of numerous modern forms see Donkin's *Text-book of Gas, Oil, and Air-engines*. Also Witz, *Traité des moteurs à Gaz et à Pétrole*.

[2] Slade, *Journ. Franklin Inst.* 1866.

exceedingly large proportion to the whole heat. The efficiency was comparatively low both on this account and on account of the limited range through which the heated products of combustion were allowed to expand. About 95 cubic feet of gas were used per horse-power-hour, which is four or five times as much as a good modern gas-engine consumes. Hugon's engine, introduced five years later, was a non-compressive engine very similar to Lenoir's. A novel feature in it was the injection of a jet of cold water to keep the cylinder from becoming too hot. These engines are now obsolete; the type they belonged to, in which the mixture is not compressed before explosion, is now represented by one small engine—Bischoff's—the mechanical simplicity of which atones for its comparatively wasteful action in certain cases where but little power is required.

252. Otto and Langen's Atmospheric Gas-engine. In 1866 Otto and Langen introduced a curious engine[1], which, as to economy of gas, was distinctly superior to its predecessors. Like them it did not use compression. The explosion occurred early in the stroke, in a vertical cylinder, under a piston which was free to rise without doing work on the engine shaft. The piston rose with great velocity, so that the expansion was much more nearly adiabatic than in earlier engines, and the ratio of expansion was greater. After the piston had reached the top of its range the gases became further cooled by giving up heat to the walls of the cylinder, and, their pressure being below that of the atmosphere, the piston came down, this time in gear with the shaft, and doing work upon it. The burnt gases were discharged during the last part of the down-stroke. A friction-coupling allowed the piston to be automatically thrown out of gear with the shaft when rising, and into gear when descending. This "atmospheric" gas-engine used about 40 cubic feet of gas per horse-power-hour, and came into somewhat extensive use in spite of its noisy and spasmodic action. After a few years it was displaced by a greatly improved type, in which the direct action of Lenoir's engine was restored, but the gases were compressed before ignition.

253. The four-stroke cycle of Beau de Rochas and Otto. The advantage of compressing the explosive mixture

[1] *Proc. Inst. Mech. Eng.*, 1875.

before igniting it in order to make the subsequent expansion large, appears to have been first clearly recognised by Beau de Rochas in a French patent of 1862. He pointed out that compression might be carried to any extent short of that which would cause the mixed gas to explode in consequence of its elevation of temperature. He further suggested a means of compressing the explosive mixture without using a separate compressing pump. His plan was to have the following four operations take place, on one side of the working piston, during four successive strokes or two revolutions of the crank-shaft.

(1) Drawing in the charge of gas and air during one whole stroke of the piston.
(2) Compression during the return stroke (into a comparatively large clearance space below the piston).
(3) Ignition at the dead point, followed by expansion during the third stroke.
(4) Discharge of the burnt gases from the cylinder during the fourth and last stroke.

This was the earliest account of the "four-stroke" cycle of operations which is now used in almost all gas-engines. Beau de Rochas further pointed out that, besides compression, high speed and small cylinder surface were conditions to be aimed at as favourable to economy. Extremely valuable as were the suggestions contained in his patent they were for a long time unproductive. It was not till 1876 that Dr Otto, who had re-invented the Beau de Rochas four-stroke cycle, introduced the highly successful gas-engine in which this action is carried out. The Otto "silent" engine (so called to distinguish it from its noisy predecessor, the engine of Otto and Langen) not only was the first gas-engine to come widely into use, but has formed the model to which, since the expiry of Otto's master patent, other gas-engines are for the most part indebted for the chief features of their action. The manufacture of the Otto engine in England by Messrs Crossley led to its rapid introduction in thousands of cases where the greater cost of gas fuel, as compared with the cost of fuel in a steam-engine, was more than balanced by the greater convenience and economy in respect of attendance of the new motor. It should be added that illuminating coal-gas—the usual fuel of these engines—is a more costly fuel than there is any need to use in a gas-engine, and is

in fact used only because it is readily obtainable. Much cheaper combustible gases, destitute of illuminating power, will serve the purposes of the gas-engine; and when gas-engine power is used on a large scale it is worth while to put down the plant necessary for the manufacture of cheap gas. This, in fact, is often done, and under such conditions the cost of fuel in the gas-engine compares favourably with the cost of fuel in the steam-engine.

254. The Otto Engine. In the Otto engine, and in many other forms which resemble it, the cylinder is single acting, with a trunk piston, and the explosive mixture is compressed before ignition into a large clearance space at the back end of the cylinder. The volume of the clearance depends on the amount of compression which is desired. In the early forms of Otto engine it was generally more than half the volume through which the piston sweeps, but is now less than half that volume. To complete the action requires two revolutions of the crank-shaft.

During the first forward stroke of the cycle gas and air are drawn in by the piston. During the first back-stroke the mixture is compressed into the clearance space. The mixture is then ignited as the crank reaches the dead point, and the second forward stroke (which is the only working stroke in the cycle) is performed under the pressure of the heated products of combustion. During the second back-stroke the products are discharged into the atmosphere through an exhaust valve, with the exception of so much as remains in the clearance space, which (except where special means are taken to remove it) is allowed to dilute the explosive mixture in the next cycle. The cylinder is kept cool enough to admit of lubrication, by means of a jacket through which a continuous circulation of water is kept up. The admission and exhaust valves are worked by a lay-shaft which is geared to run at half the speed of the crank-shaft, so that their period is double that of the piston. The same shaft serves to work the valve which determines the ignition of the explosive mixture.

255. Ignition in Gas-engines. In early forms of the Otto engine the ignition of the compressed gases was effected by carrying a flame, through a narrow port in a slide-valve, from a gas jet that was kept burning outside to the mixture within. To prevent the gases from blowing back through the valve when the explosion took place the slide was arranged so that the port in it

which served as a vehicle for the flame had passed under a cover which shut it off from the atmosphere, before it reached the fixed port on the cylinder cover through which the flame passed in to ignite the contents of the cylinder. This mode of igniting the gases is now generally abandoned and an ignition tube is used instead. This is a small closed tube of metal or porcelain which is maintained at a bright red heat by a flame playing on its outside surface, while a portion of the explosive mixture is allowed to enter the tube from the cylinder at the time when it should be fired. In most cases the ignition tube is used in conjunction with a "timing valve" which determines the instant at which the explosion occurs by being opened to allow a portion of the compressed explosive mixture to enter the ignition tube, but in some gas-engines this "timing valve" is dispensed with and the ignition tube is in free communication with the cylinder throughout, the contents becoming fired only when their pressure is raised by the back-stroke of the piston, an arrangement which tends to make the instant of ignition rather uncertain. In a few gas-engines electrical means of firing are retained: thus in the "Simplex," a successful French engine, a continuous stream of sparks is kept up, generated by a battery and induction coil in a small chamber in the cover of a slide-valve which serves as a timing valve to let the sparks take effect at the proper time, by giving the explosive mixture access to the spark chamber.

256. **Governing of Gas-engines.** The speed of a gas-engine is usually regulated by a centrifugal governor, which cuts off the supply of gas when the speed exceeds a certain limit, making the engine miss one or more explosions. The governor determines whether the gas valve shall or shall not be opened, by means of a "hit and miss" arrangement. A cam fixed on the lay-shaft, so that it makes one revolution for every two revolutions of the engine, opens the gas-admission valve by acting on a lever through an intermediate roller. This intermediate roller is carried by an arm which is caused to move sideways by the governor, in such a manner that when the speed exceeds a certain value the roller is removed and consequently the cam fails to act on the lever, and the admission valve remains closed. In some instances a stepped cam is used, giving admission to various amounts of gas corresponding with various positions of the centrifugal governor, with

the effect that the explosive mixture is weakened when the speed rises. The tendency, however, to miss fire with weak mixtures is an obstacle to this method of regulating, and more generally the gas is freely admitted when the speed is below the limit, and completely cut off when the limit is passed. In some gas-engines the inertia of a reciprocating piece is used instead of the inertia of revolving pieces to determine the admission or non-admission of gas. When the speed exceeds a limit the acceleration of the oscillating piece becomes sufficient to displace it in such a way that the gas-admission valve misses the stroke.

257. **Other Gas-engines.** The Otto, or Beau de Rochas cycle, is now so generally adopted that comparatively little interest attaches to the modes of action introduced in other types of gas-engine, which after attaining some vogue have for the most part failed to maintain their position. Mr Dugald Clerk, whose experiments have been of great service in clearing up disputed points in gas-engine theory, introduced in 1880 a motor in which the explosion took place at each forward stroke of the piston, instead of at each alternate forward stroke as in Otto's. The gas and air were inhaled by an auxiliary piston in a separate cylinder, from which they were delivered to the main cylinder just after the main piston had completed its working stroke. They entered through a trumpet mouth or large cone forming the cover at the back end of the cylinder, which had the effect of removing the kinetic energy of the stream, and hence of allowing the fresh gases to enter without intermingling much with the products of combustion already in the cylinder. The fresh charge drove the products of combustion in front of it, causing these to be expelled at exhaust ports in the side of the cylinder close to the front end of the stroke. The piston, returning, closed these exhaust ports and compressed the fresh mixture, which was ignited as usual when its compression was completed by the piston passing its dead-point at the back end[1]. The indicator diagram was almost identical with that given by the Otto engine.

It is a defect of the ordinary Otto cycle that the ratio in which the gases are expanded after ignition is no greater than the ratio

[1] See Mr Clerk's book, "*The Gas-Engine*," 1886.

in which the explosive mixture is compressed. A larger ratio of expansion is desirable, for the temperature and the pressure are still high when release occurs. The ingenious "differential" engine of Mr Atkinson (1885) was an attempt to escape this drawback. Its working chamber consists of the space between two pistons working in one cylinder. During exhaust the pistons came close together; they receded from each other to take in a fresh charge; they approached for compression; and finally they receded again very rapidly and farther than before, after ignition of the mixture, thus giving a comparatively large ratio of expansion with the further advantage that the working stroke took place so quickly as to give the burning gases comparatively little time to give up heat by conduction to the metal. At the same time, by moving bodily along through the cylinder, the pistons uncovered admission and exhaust ports in the sides, as well as an ignition tube which was kept permanently incandescent. The pistons were connected to a single crank-pin through a pair of beams or levers with connecting links at each end; this had the important disadvantage of introducing a large number of working joints.

A year or so later Mr Atkinson abandoned the use of two pistons, but succeeded in giving a single piston long and short strokes alternately, by connecting it to the crank-pin through a species of toggle joint which made two complete oscillations for each revolution of the shaft. The first oscillation served to inhale and compress the gases; the second, the amplitude of which was about twice as great, served for the working and exhausting stroke. Competitive tests made under the auspices of the Society of Arts showed that the "cycle" engine, as this form was called, had an exceptionally high efficiency, but the mechanical complication of the toggle with its still numerous joints has stood in the way of its success.

In engines made by Messrs Dick, Kerr and Co., a double-acting stroke has been used, with explosions on both sides of the piston alternately. In the original form of this engine (the "Griffin") there was what is called a scavenging stroke (or rather two strokes) in addition to the essential parts of the Otto cycle. After the products of combustion had been expelled in the fourth stroke of the ordinary cycle a fifth and sixth stroke were occupied in drawing in and expelling air from the cylinder, to sweep it clear

of the products of combustion before the next charge should be admitted. But in more recent forms of double-acting engine the scavenging strokes are omitted and the usual four-stroke Otto cycle is adhered to without change, the result of the double action being that two impulses are secured in every two revolutions of the shaft.[1] Gas-engines have been made by these makers to indicate as much as 600 horse-power.

In one or two other forms of gas-engine the front or idle side of the piston is utilized to serve as a displacer, acting like the separate displacer of the Clerk engine to inhale a mixture of gas and air into a chamber which in some cases encloses the connecting-rod and crank. The mixture then passes to the back or working end, while the burnt products of the previous stroke are driven out before it, and is then compressed and exploded; the result being that an explosion is secured at each instead of each alternate revolution. The cycle is substantially Otto's, but with its chief mechanical imperfection removed, namely, the idle revolution.

258. Action in the Cylinder of the Otto Engine. If the explosion of a gaseous mixture were practically instantaneous, producing at once all the heat due to the chemical reaction, and if the expansion and compression were adiabatic, the theoretical indicator diagram of an engine of the Otto type would have the form shown in fig. 195. OA represents the volume of clearance; AB is the admission, at atmospheric pressure; BC is the compression (which is assumed in the sketch to be adiabatic); CD is the rise of pressure caused by explosion; DE is adiabatic expansion during the working stroke; and EBA is the exhaust. The height of the point D above C may be calculated when we know the temperature at C (an element of considerable uncertainty in practice), as well as the specific heat (at constant volume) of the burnt mixture, the amount of heat evolved by the explosion, and the change of specific density due to the change of chemical constitution which explosion brings about. With the proportion of coal-gas and air ordinarily employed the change of

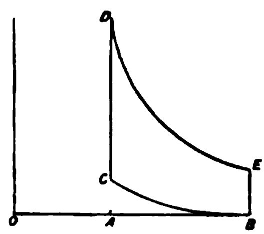


FIG. 195.

[1] For particulars of double-acting gas-engines used in the electric lighting of Belfast, see *Proc. Inst. Mech. Eng.*, July 1896.

specific density may generally be neglected, as the volume of the products would differ by less than 2 per cent. from the volume of the mixture before explosion if both were reduced to the same pressure and temperature.

The rise of pressure which actually takes place in the indicator diagrams of gas-engines when the mixture is ignited is found to be in all cases much less than the calculated rise of pressure which would be caused by a strictly instantaneous explosion. An actual diagram from an Otto engine working in its normal manner is given in fig. 196, where the reference letters distinguish the parts of a complete cycle, as in fig. 195. It shows a rapid rise of pressure on explosion, so rapid that the

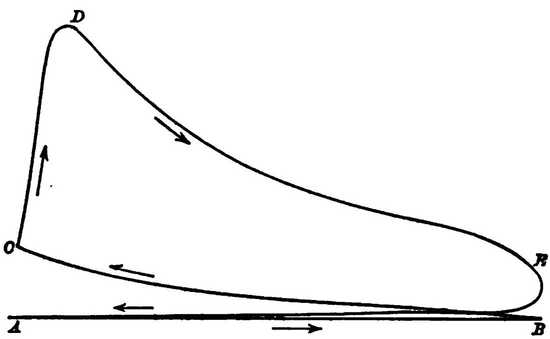


FIG. 196. Indicator diagram from an Otto Gas-engine.

volume has not very materially altered when the maximum of pressure is reached; and the specific heat at constant volume may therefore be used without serious error in calculating the amount of heat which this rise accounts for. When this calculation is made[1], it turns out that only about 60 or 70 per cent. of the potential heat of combustion in the mixture is required to produce the rise of temperature which corresponds to the rise of pressure to the highest point of the curve, namely from C to D. The remainder of the heat continues to be slowly evolved during the subsequent expansion of the hot gases. The process of combustion—a term evidently more appropriate than explosion—is essentially gradual; when ignition takes place it begins rapidly, but it continues to go

[1] See two papers by Mr Dugald Clerk, "On the Theory of the Gas-Engine," and "On the Explosion of Homogeneous Gaseous Mixtures," *Min. Proc. Inst. C. E.*, 1882 and 1886.

on at a diminishing rate throughout the whole or nearly the whole of the stroke. That part which takes place after the maximum pressure is passed constitutes the phenomenon of "after-burning" to which allusion has been made above.

259. **After-burning.** The existence of after-burning is proved not only by the fact that the maximum pressure after ignition is much less than it would be if combustion were then complete, but also by the form which the curve of subsequent expansion takes. During expansion the gases are losing much heat by conduction through the cylinder walls. The water-jacket absorbs about 40 or in some cases even 50 per cent. of the whole heat developed in the engine[1], and the greater part of this is of course taken up from the gases during the working stroke. Notwithstanding this loss of heat, the curve of expansion does not fall much below the adiabatic curve; in some tests indeed it has been found to lie higher than the adiabatic curve. This shows that the loss to the sides of the cylinder is being made up by continued developement of heat within the gas. The process of combustion is especially protracted when the explosive mixture is weak in gas; the point of maximum pressure then comes late in the

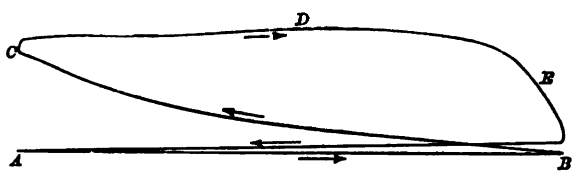


Fig. 197. Otto Engine diagram with weak explosive mixture.

stroke; and it is probable that the products which are discharged into the exhaust contain some incompletely burnt fuel. Fig. 197 is the indicator diagram of an Otto engine supplied with a mixture containing an exceptionally large proportion of air: it exhibits well the very gradual character of the combustion in such a case.

The process of explosive combustion has been examined in the experiments already referred to of Mr Clerk, who exploded mixtures of gas and air, and also mixtures of hydrogen and air, in a closed

[1] Clerk, *loc. cit.* Also, Brooks and Steward, *Van Nostrand's Eng. Mag.*, 1883; Ayrton and Perry, *Phil. Mag.*, July 1884; Slaby, Report quoted in F. Jenkin's *Lecture*, Inst. C. E., 1884, and other trials quoted by Mr Donkin.

vessel furnished with an apparatus for recording the time-rate of variation of pressure. In these experiments the pressure fell after the explosion only on account of the cooling action of the containing walls. The temperature before ignition being known, it became possible to calculate from the diagrams of pressure the highest temperature that was reached during combustion (on the assumption that the specific heat of the gases remained unchanged at high temperatures), and to compare this with the temperature which would have been produced had combustion been at once complete. Mixtures of gas and air were exploded, the proportion of gas varying from $\frac{1}{16}$ to $\frac{1}{5}$, and the highest temperature produced was generally a little more than half that which would have been reached by instantaneous combustion of the mixture. With the best proportion of coal-gas to air (1 to 6 or 7) the pressure reached its maximum one-twentieth of a second after ignition, and the temperature was then 1800° C.,—instead of 3800° C., which would have been the temperature had all the heat been at once evolved. With the weakest mixtures about half a second was taken to reach a maximum of pressure, and the temperature was then 800° C., instead of 1800° C. In this case, however, the degree of completeness of the combustion is not fairly shown by a comparison of these temperatures, since much cooling occurred during the relatively long interval that preceded the instant of greatest pressure.

To explain the phenomenon of after-burning or delayed combustion, it has been supposed that the high temperature to which the gases are raised in the first stages of the explosion prevents union from being completed,—just as high temperature would dissociate the burnt gases were they already in chemical union,—until the fall of temperature by expansion and by the cooling action of the cylinder walls allows the process of union to go on. The maximum temperature attained in the gas-engine is high enough to cause a perceptible amount of dissociation of the burnt products; it may therefore be admitted that this explanation of delayed combustion is to some extent valid. On the other hand, the phenomenon is most noticeable with mixtures weak in gas, in which the maximum temperature reached is low, and the dissociation effect is correspondingly small. It appears, therefore, that dissociation is not the main cause of the action; apart from it the process of combustion of a gaseous mixture is gradual,

beginning fast and going on at a continuously-diminishing rate as the combustible mixture becomes more and more diluted by the portions already burnt. If the mixture is much diluted to begin with, the process is comparatively slow from the first.

Much stress was at one time laid by Otto on the desirability of having a stratified mixture of gases in the cylinder, with a part rich in gas near the ignition port and a greater proportion of residual products or air near the piston. It has even been supposed that stratification of the gases is the cause of their gradual combustion. Mr Clerk's experiments are conclusive against this; the mixtures he used, which gave in some cases very gradual explosions, were allowed to stand long enough to become sensibly homogeneous. In dealing with weak mixtures it is no doubt of advantage to have a small quantity richer in fuel than the average close to the igniting port to start the ignition of the rest,—but beyond this stratification has probably no value. In the ordinary working of a gas-engine it is evident that no general stratification can occur, when account is taken of the commotion which the air and gas cause as they rush into the cylinder at a speed exceeding that of an express train, except in cases where special precautions are taken to deprive the gases of kinetic energy on their entry, as in Clerk's engine, where the gases were reduced to stillness by means of an expanding cone in order that the fresh charge should not mix with the products of the previous explosion.

260. Particulars of the Crossley-Otto Engine. Fig. 198 gives a view in plan of a Crossley-Otto engine, showing a cylinder and piston in section. The cylinder is in the form of a liner inside an outer casing, the space between the two serving as the water-jacket. The piston is of the trunk type, and is extended forward a long way with the effect that no outside guide is required. The lay shaft at the side is driven from the main shaft by skew gear wheels, and a second fly-wheel is often added outside these gear wheels in cases where uniform motion is specially desirable, as in gas-engines used to drive dynamos for electric lighting. In this instance the clearance space is 40 per cent. of the volume swept through by the piston in each stroke. The pocket at the side of the clearance space contains the exhaust valve, which is a horizontal disc valve pulled down on its seat by a

spring and lifted periodically by a lever worked by one of the cams on the lay shaft. Fig. 199 gives a vertical section through

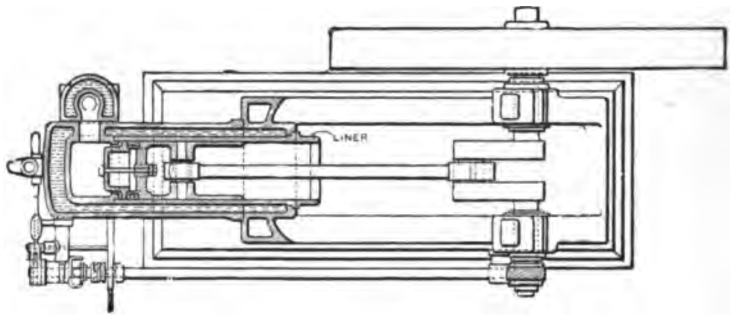


FIG. 198. Crossley-Otto Gas-engine.

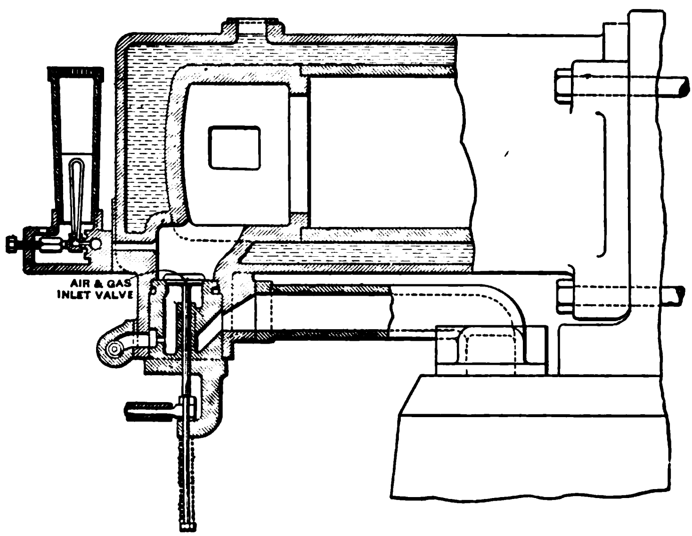


FIG. 199. Vertical section through end of cylinder of Crossley-Otto Engine.

the end of the cylinder, showing the air and gas admission valve and also the ignition tube, which is protected by an outer tube open at the top and is kept hot by a Bunsen flame burning in the space between the two. The cylinder of this engine has a diameter of $9\frac{1}{2}$ inches, and the stroke is 18 inches. As tested by Mr Clerk[1],

[1] D. Clerk, "Recent Developments in Gas-engines," *Min. Proc. Inst. C. E.*, vol. cxxiv., 1896. Figs. 198 to 201 are taken from this paper, which forms an important contribution to the literature of the gas-engine.

at a speed of 160 revolutions per minute it developed 19·25 I.-H.-P. and 15·75 B.-H.-P., with a gas consumption of 21·2 cubic feet per I.-H.-P. hour. The pressure was 48 lbs. per square inch after compression, and 200 lbs. after explosion. The mean pressure during the working stroke was 81½ lbs. The indicator diagram is given in fig. 200.

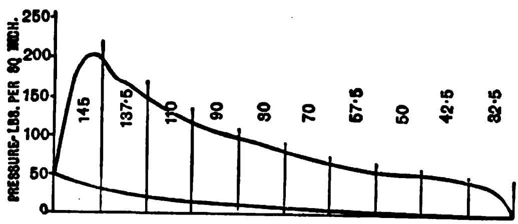


FIG. 200. Indicator diagram of Crossley-Otto Engine (1892).

In their larger engines, some of which develope 250 H.-P., Messrs Crossley use two cylinders facing one another on opposite sides of the crank, with the two connecting rods working on the same crank-pin. In a gas-engine of 400 H.-P. built by Messrs Andrews the two cylinders are placed in tandem on the same side of the shaft. The front piston has no tail-rod, but is connected with the piston behind it by two side rods, one on either side of the front cylinder. The explosions in the two cylinders take place alternately, so that one occurs in every revolution when the engine is working at full power.

In all engines using the Otto cycle with equal expansion and compression strokes, the expansion of the gases after combustion is far from complete, and then pressure at release is still comparatively high. The noise in the exhaust pipe due to this is generally mitigated by using what is called a "silencer," which is a cast-iron vessel, placed at any convenient point on the pipe, into which the gases expand before they are allowed to escape to the open air.

261. Scavenging. In most instances a gas-engine working on the Otto cycle keeps its clearance space at the end of each exhaust stroke full of unrejected products of combustion, and these mix with the incoming gas and air of the succeeding charge.

"Scavenging," or sweeping out the burnt gases from the clearance space, has occasionally been effected by a separate air-pump or (as was mentioned in § 257) by adding a pair of idle strokes to the four-stroke cycle of Otto. An ingenious and simple means of getting rid of the burnt gases is now applied by Messrs Crossley to many of their engines, with marked advantage to the efficiency of the action. This is the invention of Mr Atkinson, whose plan is to provide the engine with a rather long exhaust pipe and to open the air-admission valve a little sooner than the gas valve is opened, and a little before the end of the exhaust stroke. The exhaust gases are discharged through the exhaust pipe with a rather violent puff after each explosion, and the energy of their motion in the long pipe causes the puff to be followed by a partial vacuum. Hence when the air-admission valve is opened there is a rush of fresh air into the clearance space which sweeps before it the remainder of the burnt gases and leaves the clearance full of pure or nearly pure air. The length of the exhaust pipe is so chosen that the partial vacuum, or phase of low pressure at the end of the pipe next the cylinder, comes when the piston is nearing the end of its stroke. Mr Atkinson's invention allows scavenging to be effected without adding in any way to the working parts of the engine, and without the reduction of power that is involved by adding idle strokes to the four-stroke cycle. With an exhaust pipe about 65 ft. long, the pressure in the cylinder, in an example tested by Mr Clerk, fell towards the end of the stroke to about 2 lbs. below the pressure of the atmosphere, thereby causing a strong current of fresh air to sweep through the clearance space when the air-admission valve was opened. The action of the scavenging device is conveniently examined by causing a diagram of the exhaust stroke to be drawn by an indicator furnished with a light spring and fitted with a stop to prevent more than a small movement. Scavenging appears in all cases to reduce considerably the consumption of gas, and engines which use it show the greatest economy of fuel yet attained. It is of special advantage when a comparatively weak gas such as Dowson's is used, for it diminishes the chance of miss-fire to which weak gas is liable. Again, in large engines the presence of hot burnt gas in the mixture is apt to induce premature explosion, and this scavenging does much to prevent.

[1] *Loc. cit.* p. 111.

262. Influence of Compression. The advantage of compressing the mixture before explosion is partly mechanical and partly thermodynamic. Apart from considerations of efficiency it augments the power of the engine (relatively to its size) by causing a larger quantity of gas to enter into the action, so that a larger amount of heat is generated at each explosion and the mean pressure during the working stroke is increased. In a Crossley-Otto engine of 1881 the mixture was compressed until its pressure was about 30 lbs. per square inch; on explosion the pressure rose to 120 lbs., and the mean pressure during the working stroke was 55 lbs.[1]. In an engine of 1892 the pressure had the higher values stated in § 260. In an engine of 1894 the compression was carried to 75 lbs., the pressure on explosion was 318 lbs., and the mean pressure during the working stroke was 113½ lbs.[1]. Compared with the engine of 1881 this last engine, in consequence of the greater compression of its charge, did more than twice as much work per cubic foot swept through by the piston.

But apart from the increase in power, compression produces an increase in efficiency. The higher pressure reached on explosion allows a greater range of expansion to follow without causing the pressure to fall unduly low. With the same terminal pressure in both cases increased compression enlarges the indicator diagram for the same charge of gas. Further, the reduced volume of the clearance implies a reduced surface for the cooling of the gas. And it may be added that, in engines which do not use scavenging, the diminished clearance means a diminished quantity of residual gas to mix with the succeeding charge. The absence of burnt gas from the mixture contributes to efficiency, and the greater is the compression the less is the volume of the residual products.

Comparing Crossley-Otto engines of similar size but of different dates, with the object of showing the gain in efficiency which has resulted from increased compression, Mr Clerk cites the following results of tests:—

Date.	Pressure at end of compression above atmosphere. lbs. per sq. inch.	Efficiency, or fraction of heat in gas converted into indicated work.
1882–8	38·0	0·16
1888–94	66·0	0·19
1894	87·5	0·25

[1] Clerk, *loc. cit.*

The consumption of gas per I.-H.-P. hour was reduced from 24 cubic feet in the first of these cases to 14·8 cubic feet in the last.

This improvement is to be ascribed mainly to the greater compression used in modern engines, but some of it is due to increased piston speed, giving a more nearly adiabatic expansion; and probably a considerable part of the final gain is due to "scavenging," which was used in the last of these three examples.

The increased compression which is a feature of modern gas-engines has been made possible by the substitution of the ignition tube for the old ignition slide-valve. The slide-valve was held against the end of the cylinder by springs, which had to exert enough force to keep it from being blown off the face when the explosion occurred. So long as this arrangement was adhered to the pressure reached during the explosion could not be allowed to exceed a very moderate limit. With the abolition of the slide-valve this limit disappeared and high compression became practicable.

263. Self-starting appliance for Gas-engines. Small gas-engines are readily enough started by hand by pulling the fly-wheel round until explosion begins to happen, but engines of large power have to be provided with some safer and less cumbrous means of being set in motion. In Mr Clerk's starter, which is used by Messrs Crossley, the object is attained of filling the clearance space in the cylinder with a mixture of gas and air under pressure, so that the engine may start with a high-pressure explosion. The mixture, however, is not pumped in, but is driven in by the combustion of gas in a separate vessel into which the mixture is originally introduced at atmospheric pressure. The arrangement is shown in fig. 201. *D* is the separate vessel into which gas and air are admitted, but not under pressure. The igniter *T* fires this mixture, which begins to explode at the top of *D*. The explosion drives the lower portion of the mixture over into the engine cylinder *A* through the check-valve *B*, and it accumulates there under pressure. Finally the flame passes the check-valve and ignites the compressed mixture behind the piston, thereby giving the impulse required for the start, the crank having been set beforehand some way forward from its dead-point.

The Lanchester low-pressure starter causes gas first to pass into the space behind the piston and to mix with the air there. Part of the mixture is allowed to escape through a cock of special form,

where it is ignited. When the cock is closed the flame travels back into the cylinder and the mixture there explodes, giving the

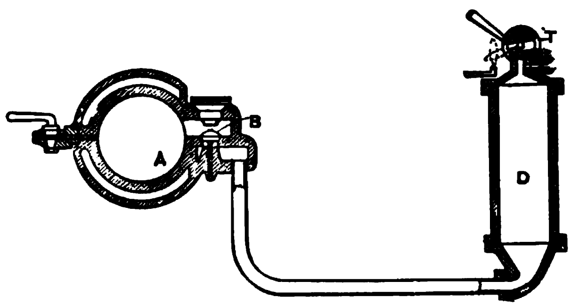


FIG. 201. Clerk's Pressure Starter for Gas-engines.

necessary impulse to the piston. In other cases the starting is effected by pumping a charge into the cylinder, or by admitting compressed air from a reservoir after the clearance space has already been filled with gas, and then firing the mixture.

264. Performance of Gas-engines. In small gas-engines of the Otto type using illuminating coal-gas as fuel the consumption is now reduced to something less than 20 cubic feet per indicated horse-power-hour when working at full power. In large engines it may be as low as 15 cubic feet or even less. The mechanical efficiency is usually between 0·80 and 0·85; in other words, 20 cubic feet per hour per I.-H.-P. corresponds to say 23 or 25 cubic feet per brake H.-P.

Coal-gas has a heating power which ranges from about 480,000 to 620,000 foot-pounds per cubic foot. If we take 520,000 as a representative average value, the heat which is equivalent to one H.-P. hour would be given by the consumption of 3·8 cubic feet of gas. Hence an engine which consumes 19 cubic feet per I.-H.-P. hour has a thermodynamic efficiency of 0·2, that is to say, it converts just 20 per cent. of the energy of the fuel into work. Efficiencies as high as 0·25 and even 0·27 have been found in trials of gas-engines using high compression and "scavenging." We have seen in Chapter V. that the most efficient steam-engine converts only about 15 per cent. of the energy of the fuel into work, and in steam-engines that are small enough to be fairly compared with

the gas-engines to which these figures refer, the fraction converted is rarely more than 10 per cent. The superiority of gas-engines over steam-engines, from the thermodynamic point of view, is therefore considerable: it is of course due to the greater range of temperature through which the working substance is carried, and especially to the high mean temperature of the working substance during which heat is being taken in, or rather being developed by the combustion of the substance itself, a process which corresponds to the reception of heat by the working substance of an external combustion engine.

The following figures are quoted from a Report of trials conducted by the Society of Arts in 1888[1]. They relate to three tests of a Crossley-Otto engine with the moderate compression which was then usual, which was one of the three gas-engines submitted for trial. One of the others, the Atkinson "cycle" engine, which was referred to above as having a long expansive stroke, had a rather higher efficiency, since it consumed only 19·2 cubic feet

TABLE XIV.—*Trials of a Crossley-Otto Gas-engine* (Society of Arts, 1888).

Load.................... Duration of Trial ...	Full Power. 6 hours.	Half Power. 3 hours.	None. ½ hour.
Revolutions per minute	160·1	158·8	161·0
Explosions per minute	78·4	41·1	10·2
Maximum pressure	197 lbs.	196 lbs.	148 lbs.
Mean effective pressure	67·9 „	73·4 „	66·7 „
Indicated H.-P.	17·12	9·73	2·19
Brake H.-P.	14·74	7·41	——
Mechanical efficiency	0·86	0·76	——
Gas per hour (main)	351·8 c. ft.	202·6 c. ft.	49 0 c. ft.
„ „ (ignition)	3·5 „	3·2 „	
„ „ (total)	355·3 „	205·8 „	
Gas per I.-H.-P. hour (main)	20·5 „	20·8 „	
„ „ „ (total)	20·8 „	21·2 „	
Gas per B.-H.-P. hour (total)	24·1 „	27·8 „	
Cooling water per hour	713 lbs.	480 lbs.	
Rise of temperature of cooling water	71·8° F.	71·3° F.	

[1] The official report of these trials, with a further descriptive paper by Professor Kennedy (*Jour. Soc. Arts*, March 1889), should be referred to as an example of a scientifically conducted test.

of gas per I.-H.-P. hour including the supply required for the ignition. The trials quoted below were made with the engine working at full power and half power, and also running without external load.

Taking the full-power trial, it appears that of every 100 units of energy in the fuel, about 19 were turned into work, 43 were rejected in the jacket water, and 38 were rejected in the exhaust. An indicator diagram from this trial is given in fig. 202, drawn to absolute scales of pressure and volume, and with lines BC and EF

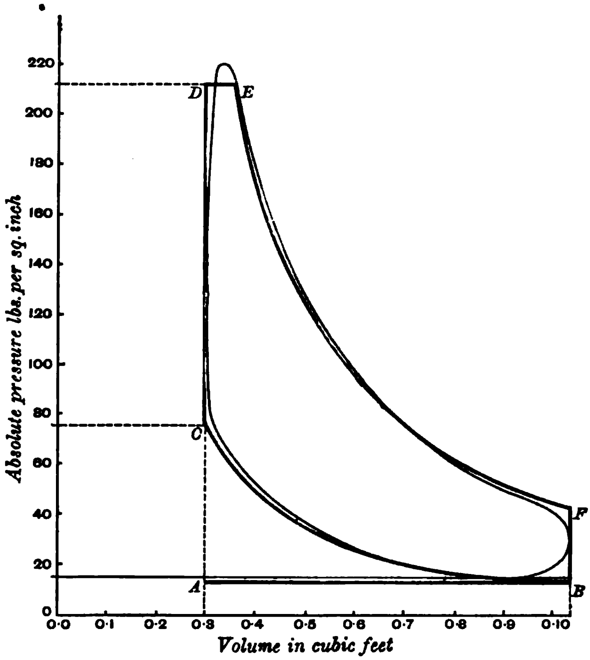


FIG. 202. Otto Engine diagram, Society of Arts Trials.

added to illustrate hypothetical compression and expansion curves of the form $PV^n = \text{const.}$ An approximation to the real compression curve is obtained when n is taken equal 1·38, and to the real expansion curve when n is taken equal to 1·435. The fact that the last index is higher than the value of γ for the mixed gases shows that heat is being received during expansion (§ 39) in consequence of after-burning, and notwithstanding the loss to the jacket which is going on more actively in this stage of the cycle than in any other. The absolute temperature is estimated

to have been approximately 3440° Fah. at E where it was a maximum, 2130° at F after expansion, and 1060° at C after compression.

As a more recent example a trial may be quoted which was made by Mr Clerk in 1894 with a small Crossley engine, using Mr Atkinson's scavenging device[1]. An indicator diagram taken in this trial is reproduced in fig. 203. The cylinder was 7 inches in

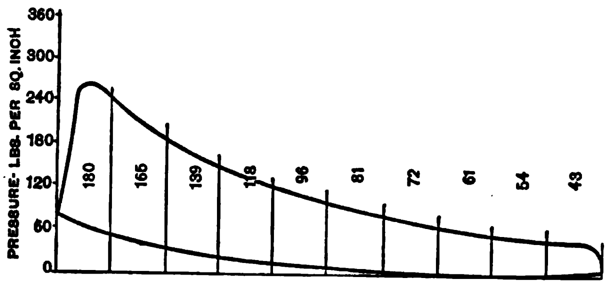


FIG. 203. Crossley Otto Scavenging Engine. (1894.)

diameter, with a 15-inch stroke, and the clearance space was 34 per cent. of the volume swept through by the piston. The engine ran at 200 revolutions per minute and developed 14 I.-H.-P. with a mean pressure in the working stroke of 100 lbs. The highest pressures were 87·5 lbs. in compression and 275 lbs. in explosion. The consumption of gas (estimated to have a heating value of 530,000 foot-pounds per cubic foot) was 14·8 cubic feet per I.-H.-P. hour and 17 cubic feet per B.-H.-P. hour. This makes the thermodynamic efficiency equal to 0·25, which is a remarkably good result for so small an engine.

When a gas-engine is run at less than its full power the consumption of gas is of course reduced, but not in simple proportion to the reduction of load. In general for an engine running at one speed the total gas used per hour is at least very approximately proportional to the brake-horse-power plus a constant, the constant being the quantity of gas consumed per hour when the engine is running without any external load. A line drawn, like the Willans' line for a steam-engine, to show the total gas used per hour in relation to the brake-horse-power is nearly straight.

[1] *Min. Proc. Inst. C. E.*, vol. cxxiv. p. 102.

Fig. 204 shows this line drawn for the three trials quoted in Table XIV, the ordinates taken being the total gas used per hour in the

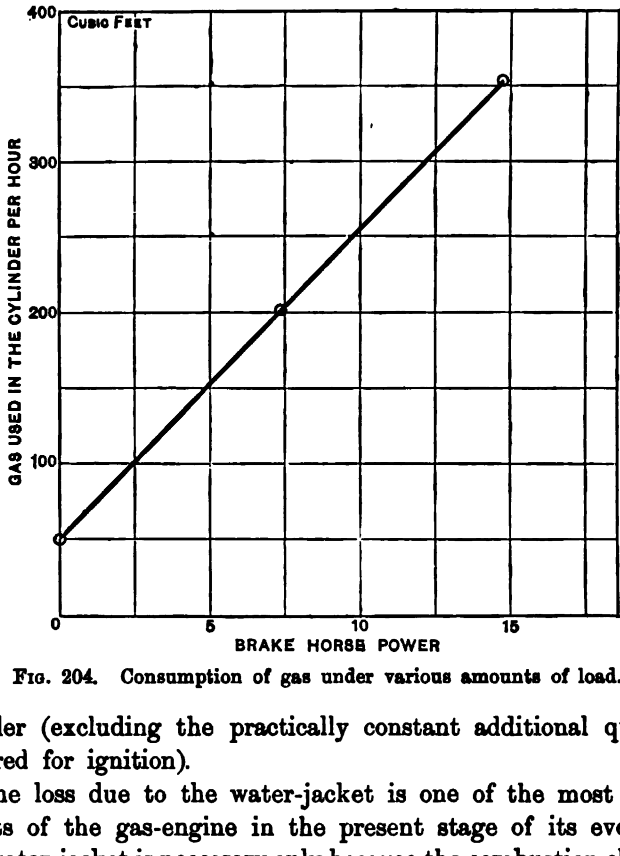


FIG. 204. Consumption of gas under various amounts of load.

cylinder (excluding the practically constant additional quantity required for ignition).

The loss due to the water-jacket is one of the most serious defects of the gas-engine in the present stage of its evolution. The water-jacket is necessary only because the combustion chamber and the working cylinder are one: if it were found practicable to separate them, by interposing a cushion of idle air to prevent the hot products of combustion from reaching the working surface this source of loss might possibly be removed. Excessive loss also results from the high temperature at which the gases are discharged. A partial but important remedy for this is to be sought in extending the expansion, as Mr Atkinson did in his "cycle" engine; but the most complete cure is perhaps to be looked for in the application to gas-engines of the regenerator of Stirling. Though attempts in this direction have already been made, the regenerative gas-engine still awaits development.

265. Ideal performance of an internal-combustion engine. That there is an immense margin for improvement in the efficiency is clear from the consideration that the gas-engine, efficient as it is, falls much more short of the efficiency that is theoretically attainable than does its older rival, the steam-engine. In estimating the ideal maximum we may take as the lower limit the temperature of the atmosphere, or say 520° Fah. (absolute). The trials which have been quoted agree with others in showing that an absolute temperature of about 3440° Fah. is reached in the combustion of coal-gas under conditions such as obtain in gas-engines. If all the heat were generated at this temperature the formula

$$\frac{\tau_1 - \tau_2}{\tau_1}$$

would be applicable as a measure of the ideal efficiency, to be approached, as has just been indicated, by avoiding jacket losses and by using a regenerator to assist in making the cycle reversible throughout. The value which this fraction takes, with the extremes of temperature named above, is 0·85.

But in order to bring this ideal within reach the gas and air would have to be heated separately (by compression or by the use of the regenerator) to the maximum temperature before combustion was allowed to begin, and would have to be prevented from cooling until combustion was complete. This would imply an entirely different kind of action from that which exists in the Otto or any other existing type of engine, where the generation of heat is necessarily associated with a rise of temperature. It will therefore be a fairer comparison if we take as the ideal standard of performance that of an engine in which the combustion goes on between two defined temperatures, a lower initial and a higher final limit, and in which the action is reversible in all other respects. Calling τ_0 the temperature to which the gases are raised before ignition, τ_1 the maximum at which (in the ideal case) combustion will be supposed to be complete, and τ_2 the lower extremity of the whole range, the greatest amount of work that can be done per unit of substance is expressed by

$$W = \int_{\tau_0}^{\tau_1} \frac{dH}{\tau} (\tau - \tau_2)$$

$$= \sigma \int_{\tau_0}^{\tau_1} \frac{d\tau\,(\tau - \tau_2)}{\tau},$$

if the specific heat during combustion is assumed to be constant and equal to σ. This makes

$$W = \sigma(\tau_1 - \tau_0) - \sigma\tau_2 \log_e \frac{\tau_1}{\tau_0}$$

and the efficiency

$$\frac{W}{\sigma(\tau_1 - \tau_0)} = 1 - \frac{\tau_2}{\tau_1 - \tau_0} \log_e \frac{\tau_1}{\tau_0}.$$

Taking the numerical values as above, namely $\tau_1 = 3440$, $\tau_2 = 520$, and $\tau_0 = 1060$, this gives 0·74 as the ideal efficiency. The best results obtained in trials show that about one-third of this theoretical efficiency is realised in the performance of modern gas-engines.

266. Use of cheap gas. When the gas-engine is used to furnish power on a small scale the convenience of using ordinary coal-gas as fuel compensates for its comparatively great cost. But for the application of the gas-engine on any large scale a fuel cheaper than illuminating coal-gas is essential. In many metallurgical processes a cheap gaseous fuel is obtained by letting air pass in limited quantity through incandescent coal or coke; the gas that passes off consists of carbonic oxide formed by union of the carbon with the oxygen of the air, diluted by the nitrogen which has passed through without change. This "producer gas," as it is called, is too weak to be effectually used in ordinary gas-engines. A stronger fuel, called "water gas," consisting of mixed hydrogen and carbonic oxide, is formed by blowing steam through incandescent carbon. The gas which has hitherto been most used as a substitute for common coal-gas in gas-engines has a quality intermediate between these two: it is made by Mr Dowson's process of sending a jet of mixed air and steam through a chamber which contains coke or anthracite at a red heat. The steam is formed in a small auxiliary boiler, and blows through a kind of injector or jet-pump, in which it takes up air, and the two pass together into the hot carbon chamber. From it the gas (consisting of a mixture of about 25 per cent. of carbonic oxide with 20 per cent. of hydrogen, and the rest nitrogen) is led through "scrubbers" into a holder which feeds the engine. The whole plant can be erected close to the engine, takes up no great space and requires little attention. The gas obtained by this means has only about one-fourth of the calorific value of ordinary coal-gas. But by

restricting the proportion of air admitted to the cylinder and compressing the mixture more strongly than is necessary in the case of ordinary coal-gas, the mixture is found to ignite well, developing a high initial pressure and giving a diagram resembling that given by ordinary coal-gas[1]. An example of a diagram from an engine using Dowson gas and indicating about 100 H.-P. is reproduced from Mr Clerk's paper in fig. 205. Incidentally it

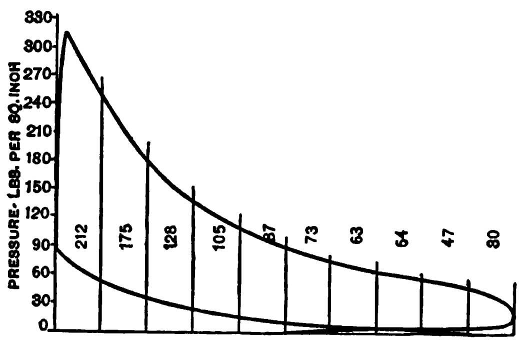


FIG. 205. Crossley-Otto Scavenging Engine with Dowson Gas.

illustrates the process of "scavenging" by showing the exhaust line cross the atmospheric line somewhere near the middle of the stroke. The makers of the "Simplex" engine employ a plant for making cheap gas somewhat similar to Mr Dowson's. A blowing fan, worked by the engine itself, sends air into the bottom of a chamber containing red-hot fuel, and at the same time a small stream of water is made to trickle upon the grate of the chamber, forming steam which rises along with the air; the combustible gases produced in this way are led off at the top through a scrubber to a gas-holder.

When a gas-engine has its fuel prepared by processes such as these a direct comparison becomes possible between its consumption and that of the steam-engine, since we have to deal in the first instance with solid fuel in both cases, supplied at a cost per lb. which if not identical is at least readily comparable. Messrs Crossley have applied Mr Dowson's process at their own

[1] For particulars of Mr Dowson's process and its application to gas-engines see his papers, *Min. Proc. Inst. C. E.*, Vols. LXXIII. LXXXIX. and CXII.

works to furnish gaseous fuel for engines developing about 300 I.-H.-P., and as a result of their experience they guarantee that their large engines will not consume more than 1 lb. of anthracite or 1¼ lb. of coke per I.-H.-P. hour. Trials in other places have established that even a small gas-engine may by this means develope a horse-power-hour by burning no more than 1 lb. of coal. A large steam-engine requires nearly twice as much, and a steam-engine small enough to make the comparison fair requires three or four times as much fuel. It is therefore not surprising that the gas-engine is every day becoming a more formidable rival to the steam-engine, for moderately great as well as for small powers. The mechanical objections, however, to the Otto cycle are more serious in large sizes than in small, and it may be conjectured that before large gas-engines become generally adopted as substitutes for steam-engines an action will have to be devised which gives a more uniform effort and wastes less of the piston's movement in idle strokes.

267. Fluctuations of speed in engines using the Otto cycle. The principles on which the fluctuations in the speed of any engine may be estimated when the indicator diagrams are given have been explained in an earlier chapter. An engine using the four-stroke cycle of Otto requires to have larger fly-wheel capacity than a steam-engine, on account of the long idle interval when no work is done on the crank. But the problem of determining the fluctuation in speed is comparatively simple, for the period when an excess of energy is being given out by the piston extends over a whole stroke. For this reason the inertia of the reciprocating parts does not have to be taken into account, and there is no need to draw a diagram of crank effort in order to determine what is the excess of energy which the fly-wheel has to absorb. An example will show best how the calculation may be made. Taking the indicator diagram of a particular gas-engine, it was found that the mean height above the atmospheric line corresponded to 88·7 lbs. per sq. inch during the firing stroke, and 15·1 lbs. during the compression stroke, and further that the mean height of the exhaust line above the suction line was 4·4 lbs. Hence the net indicated work was that which would be done by an effective pressure of $88{\cdot}7 - 15{\cdot}1 - 4{\cdot}4$ or 69·2 lbs. acting throughout one stroke (out of four). The pressure equivalent to the

mean resistance overcome by the piston was therefore $\frac{69 \cdot 2}{4}$ or 17·3 lbs. per square inch. Hence during the firing stroke the effective pressure exceeded the pressure equivalent to the mean resistance by $88 \cdot 7 - 17 \cdot 3$ or 71·4 lbs. per square inch. This excess of pressure acting throughout one stroke does the work ΔE which the fly-wheel has to absorb.

In the instance referred to the diameter of the cylinder was $7\frac{1}{2}$ inches and the stroke 9 inches, making ΔE equal to 2365 foot-pounds. The mean speed was 250 revolutions per minute, and the moment of inertia of the fly-wheels, expressed in lbs. and feet, was 8950. In the notation of § 208, Chapter X, $\omega_1 - \omega_2$, or the whole change in angular velocity between the greatest and least speed, is $\frac{\Delta E}{I\omega_0}$, which in the present example is

$$\frac{2365 \times 32 \cdot 2 \times 60}{8950 \times 250 \times 2\pi} = 0 \cdot 325.$$

Hence ΔN, the greatest change in the speed when expressed in terms of the number of revolutions per minute, was $\frac{0 \cdot 325 \times 60}{2\pi}$ or 3·1, or in round numbers one-eightieth of the mean speed. The smallness of the variation in this example is due to the high mean speed and to the use of two particularly heavy fly-wheels.

268. Oil-engines. Liquid fuel may be used in an internal-combustion engine either by evaporating it at a low temperature and allowing the vapour to pass into the engine mixed with air, to be compressed and burnt or exploded there as in gas-engines, or by injecting a liquid in the form of a jet or spray into a hot chamber where it is converted into a true gas and is then used as in a gas-engine. In early petroleum engines the oils burnt were of a readily vaporisable class, of low density, and "flashing" at a low temperature. Air was forced through the oil, or the oil was stirred up with it or sprayed into it, with the effect that the air became charged with combustible vapour. The risk that attends the storage and use of such light oils as well as their greater cost led to the design of engines in which heavier and less inflammable oils could be consumed. One of the first of these was the Brayton engine (1873), in which air was forced by a compressing pump at a pressure of about 60 lbs. per

square inch through petroleum, after which it passed through a regenerator, heated by the exhaust, and was delivered into the cylinder, where it was made to burn under nearly constant pressure as it entered, expanding in volume and causing the piston to perform the working-stroke. This method of burning (continuously at nearly constant pressure) is interesting as a possible alternative to the method usual in gas-engines of burning suddenly at nearly constant volume. The Brayton engine is said to have used 2·7 lbs. of oil per brake-horse-power hour. In more modern oil-engines using the Otto cycle the combustion has been reduced to about one-third of this amount. In their main mechanical features and in the use of a water-jacket to cool the cylinders oil-engines do not differ from gas-engines. Some of them still use oils with a low flash-point, but in most cases the oil is of the same comparatively safe kind as is supplied for use in lamps.

The oil-engine shares with the gas-engine, though in rather less degree, the advantage of requiring but little attention; it can be used in situations where there is no supply of gas, and the cost of oil per horse-power hour is less than that of town gas. For engines of large power a cheaper fuel is obtained by producing gas from coke or from coal, but as a small motor at once convenient and inexpensive the oil-engine cannot fail to take an important place.

269. Priestman's Oil-engine. One of the earliest engines to use heavy oil successfully was Priestman's, which has a separate vaporising chamber, into which oil is sprayed along with air from a closed tank. The tank is partly full of oil, and the air above it is kept at a moderate pressure by means of a small pump. Two pipes leading from the air and oil spaces respectively are united to form a spraying nozzle which delivers a stream of finely divided oil mixed with a little air into the vaporising chamber. This chamber is heated by the exhaust gases to a temperature of 200° or 300° F. Before the engine starts the vaporiser is heated by a special oil flame. The vaporised oil is drawn into the cylinder in the suction stroke, along with a large additional quantity of air, then compressed by the working piston, exploded and exhausted as in other engines of the Otto type. A little of the vapour is condensed during compression, and this serves to lubricate the

cylinder. Ignition is by an electric spark, and the governor acts by adjusting the amount of oil and air admitted to the vaporiser through the spraying nozzle. The proportion of oil to air is maintained, so that the engine has an explosion at every second revolution whether the load is heavy or light, but the energy of the explosion is varied to suit the load[1].

Professor Unwin's tests of a Priestman engine indicating eight or nine horse-power at full load showed a consumption of 0·84 lbs. per I.-H.-P. hour when the petroleum was of a quality known as "Daylight" oil, and 0·95 to 0·99 lbs. when a rather heavier oil called Russolene was used. The calorific value of the oils was about the same in both cases, but it appeared that the lighter oil vaporised more easily and completed its combustion earlier in the stroke. The following table is given for a full-power trial with Russolene as fuel.

TABLE XV. *Performance of Priestman's Oil-engine.*

Heat due to combustion of oil	100	per cent.
Work done on Brake	13·3	„
Engine friction	2·8	„
Indicated work	16·1	„
Heat rejected to jacket water	47·5	„
„ „ in exhaust gases	26·7	„

The remainder, amounting to between 9 and 10 per cent., was lost by radiation or otherwise unaccounted for. The thermodynamic efficiency (about 0·16) is not very greatly less than that of a gas-engine, and much higher than that of a steam-engine of the same size.

270. The Hornsby-Ackroyd Oil-engine. This form is remarkable for the extreme simplicity of the means adopted to effect the explosion. The oil is rendered gaseous in a small hot chamber which forms an extension of the cylinder and in which the explosion subsequently occurs. The explosions keep the chamber at a high temperature; before the engine starts it is heated by an outside oil-lamp furnished with a blowing fan. A

[1] *Min. Proc. Inst. C. E.*, Vol. CIX., 1892.

proper quantity of oil for each explosion is separately pumped into this chamber, which when the oil enters is full of the hot products of combustion from the previous stroke. The heat of the chamber converts the oil into gas, which does not at first ignite, no fresh air being present and the temperature being barely high enough in any case. Meanwhile the piston has drawn in air during the forward stroke, and when this is compressed during the next back stroke some of it enters the hot chamber; at the same time the temperature of the air is raised by compression, and the result is that the explosion takes place just as the piston passes its dead point. It is found that no special appliance such as a timing valve or an electric spark is necessary to secure that ignition shall occur at the proper place of the stroke. What is required is that the amount of compression shall be suitable. Too much compression would make the mixture ignite prematurely, before the piston reaches the end of its back-stroke; and provision is made to adjust the amount of compression, so that ignition may be neither early nor late, by shortening or lengthening the connecting rod, which has the effect of enlarging or reducing the clearance. The governor acts by controlling the delivery of oil to the chamber, which it does by opening more or less a by-pass through which a portion of the oil from the pump is allowed to escape back to the oil reservoir when the speed of the engine becomes too high. The effect is that the explosive mixture is weakened when the speed rises. An indicator diagram from one of these engines is given in fig. 206, with a scale of pressures marked at the side.

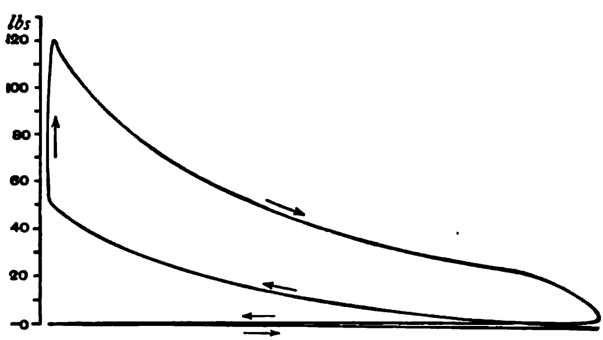


FIG. 206. Indicator diagram of Hornsby-Ackroyd Oil-engine.

271. Trials of Oil-engines by the Royal Agricultural Society[1]. Oil-engines by ten different makers were examined in these trials, all using the Otto four-stroke cycle, but differing more or less in the mode of injecting and vaporising the oil, and in the arrangement for ignition. One of the engines tested was the Hornsby-Ackroyd engine already described; another was an oil-engine by Messrs Crossley closely similar in mechanical detail to the gas-engines of the same makers. It had a separate vaporising chamber kept hot by an external oil-fed flame, an igniting tube kept hot by the same flame, and a timing valve to control the instant of the explosion. Vaporised oil was admitted to the cylinder by a valve, the action of which was intercepted by the governor when the speed became too high. The same movement which opened this valve admitted a fresh charge of oil to the vaporising chamber.

The consumption of oil in these two engines was found to be lower than in the other engines submitted for trial. The oil used was "Russolene" with a total calorific value of about 19,700 thermal units per lb. On a short full-power trial the oil consumed per hour, per brake-horse-power, was 0·82 lbs. in Messrs Crossley's engine (including the supply to the external lamp) and 0·98 in Messrs Hornsby's. The consumption per I.-H.-P. hour was also measured, but the difficulty of estimating fairly the indicated power of an oil-engine, especially when the amount of oil varies from one stroke to another, is so great that a statement of the oil used per brake-horse-power is to be preferred. On another full-power trial lasting for three days the figures were 0·90 lbs. and 0·92 lbs. respectively.

These figures would correspond to a consumption of say 0·75 lbs. per I.-H.-P. hour, and to a thermodynamic efficiency of $\frac{2545}{0{\cdot}75 \times 19700}$ or 0·17. To make these results directly comparable, however, with those stated in § 269, the calorific value of the oil would have to be reckoned in the same way as was done there, namely by allowing a rebate for the latent heat of the water vapour contained in the products of combustion. This would

[1] These trials were carried out at the Cambridge meeting of the Society in 1894, by Prof. D. S. Capper, Mr J. B. Denison, and the Author. See *Jour. Roy. Agricult. Soc.*, Dec. 1894. The results of a second series of trials of portable oil-engines will be found in the same Society's Journal for 1901.

reduce it to 18,600 thermal units and would increase the efficiency to 0·18. The cost of the oil was less than one halfpenny per B.-H.-P. hour.

272. The Diesel Motor. Mr R. Diesel has described a heat motor possessing several distinctive features, and has embodied these in an oil-engine which has achieved a remarkably high efficiency[1]. It is, in the thermodynamic sense, a defect in ordinary gas-engines and oil-engines that the combustion begins while the working substance is still comparatively cold, and consequently a part of the heat is taken in at a relatively low temperature. Mr Diesel endeavours to reach the temperature of combustion before combustion begins, by compressing the air (more or less adiabatically) to a very high degree before admitting the fuel. He then injects the fuel slowly, letting expansion go on at the same time and thus securing approximately isothermal combustion. The supply of fuel is then cut off and a prolonged expansion, more or less adiabatic in character, completes the process. The advantage of high compression, already pointed out in relation to gas-engines, is here realised to an unprecedented degree.

In the Diesel engine the main operation of air-compression, combustion, and expansion goes on in a single-acting cylinder which works in the four-stroke Otto cycle. A small supplementary air-pump delivers air at a pressure of about 50 atmospheres, which is a higher pressure than is reached in the main cylinder. By aid of this highly compressed air petroleum is injected into the clearance space, burning as it is admitted, under a pressure of between 30 and 40 atmospheres.

Tests of a 20 H.-P. engine in 1897 by Prof. Schröter and others have shown in two trials a consumption of only 0·52 and 0·54 lbs. of oil per hour per brake-horse-power, the oil having a heating value of 18,400 thermal units. As much as 34 per cent. of the heat was converted into indicated work, and 26 per cent. into work on the brake. This may fairly claim to be the highest efficiency yet recorded for any form of heat-engine.

[1] *Theorie und Construction eines rationellen Wärmemotors*, 1893. See also three papers, *Zeitschrift des Vereines deutscher Ingenieure*, July 1897, a summary of which is given by Mr Donkin in *The Engineer*, Oct. 15, 1897.

APPENDIX.

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in lbs. per sq. in. p.	Temperature. Fah. t.	Temperature. Absolute τ.	Volume of 1 lb. in cubic feet V.	Entropy. Water ϕ_w.	Entropy. Steam ϕ_s.
0·085	32	493	3416	0	2·215
0·5	80	541	640	0·095	2·052
1·0	102½	563½	332	0·136	1·985
1·5	116	577	227	0·160	1·947
2·0	126	587	173	0·177	1·922
2·5	135	596	140	0·190	1·900
3·0	142	603	118	0·202	1·885
3·5	148	609	102	0·212	1·873
4·0	153	614	90·4	0·221	1·862
4·5	158	619	80·7	0·229	1·851
5	163½	624½	73·2	0·237	1·840
6	171	632	61·7		
7	177	638	53·4		
8	183	644	47·1		
9	188	649	42·1		
10	192½	653½	38·1	0·282	1·782
11	197	658	34·9		
12	201½	662½	32·1		
13	205½	666½	29·8		
14	209½	670½	27·8		
15	213	674	26·1	0·313	1·748
16	216	677	24·5		
17	219	680	23·2		
18	222	683	21·9		
19	225	686	20·8		
20	227½	688½	19·9	0·334	1·723
21	230	691	19·0		
22	232½	693½	18·2		
23	235	696	17·5		

Pressure in lbs. per sq. in. p.	Temperature. Fah. t.	Temperature. Absolute τ.	Volume of 1 lb. in cubic feet V.	Entropy. Water ϕ_w.	Entropy. Steam ϕ_s.
24	237½	698½	16·8		
25	240	701	16·1	0·352	1·703
26	242½	703½	15·5		
27	244½	705½	15·0		
28	246½	707½	14·5		
29	248½	709½	14·0		
30	250½	711½	13·6	0·368	1·687
32	254	715	12·8		
34	257½	718½	12·1		
36	261	722	11·5		
38	264	725	10·9		
40	267	728	10·4	0·391	1·666
42	270	731	9·92		
44	273	734	9·49		
46	276	737	9·10		
48	278½	739½	8·74		
50	281	742	8·41	0·411	1·649
52	283½	744½	8·11		
54	286	747	7·83		
56	288½	749½	7·57		
58	290½	751½	1·32		
60	292½	753½	7·09	0·427	1·634
62	294½	755½	6·88		
64	297	758	6·67		
66	299	760	6·47		
68	301	762	6·28		
70	303	764	6·11	0·441	1·621
72	305	766	5·96		
74	306½	767½	5·81		
76	308½	769½	5·67		
78	310	771	5·53		
80	312	773	5·40	0·452	1·611
82	313½	774½	5·27		
84	315	776	5·15		
86	317	778	5·04		
88	318½	779½	4·94		
90	320	781	4·84	0·463	1·603
92	321½	782½	4·74		
94	323½	784½	4·64		
96	325	786	4·55		
98	236½	787½	4·47		
100	327½	788½	4·39	0·473	1·596
105	331	792	4·19		
110	334½	795½	4·01	0·482	1·589

Pressure in lbs. per sq. in. p	Temperature.		Volume of 1 lb. in cubic feet V.	Entropy.	
	Fah. t.	Absolute τ.		Water ϕ_w.	Steam ϕ_s.
115	338	799	3·85		
120	341	802	3·70	0·491	1·582
125	344	805	3·57		
130	347	808	3·44	0·499	1·575
135	350	811	3·32		
140	353	814	3·20	0·506	1·569
145	356	817	3·10		
150	358½	819½	3·01	0·513	1·564
155	360½	821½	2·92		
160	363	824	2·84	0·520	1·560
165	366	827	2·76		
170	368½	829½	2·68	0·526	1·556
175	370½	831½	2·60		
180	373	834	2·53	0·532	1·552
185	375	836	2·47		
190	377	838	2·41	0·537	1·548
195	379½	840½	2·35		
200	381½	842½	2·30	0·542	1·544
205	383½	844½	2·25		
210	385½	846½	2·20	0·547	1·540
215	387½	848½	2·15		
220	389½	850½	2·10	0·552	1·537
225	391½	852½	2·05		
230	393½	854½	2·00	0·557	1·534
235	395½	856½	1·96		
240	397½	858½	1·92	0·562	1·531
245	399	860	1·88		
250	401	862	1·85	0·567	1·528

The temperatures are stated to the nearest half degree Fahrenheit.

The volumes are calculated from Regnault's data, taking 778 foot-pounds as the value of J.

ϕ_w is the entropy of water, per lb., at the temperature at which steam would be formed under the assigned pressure, the entropy of water at 32° Fah. being used as the zero from which ϕ is reckoned.

ϕ_s is the entropy of saturated steam, per lb., reckoned from the same zero. The quantities ϕ_s and ϕ_w are connected by the equation $\phi_s - \phi_w = \frac{L}{\tau}$.

For values of H and h see Table I. p. 67.

INDEX.

CAMBRIDGE: PRINTED BY J. & C. F. CLAY, AT THE UNIVERSITY PRESS.

Куз